POWER

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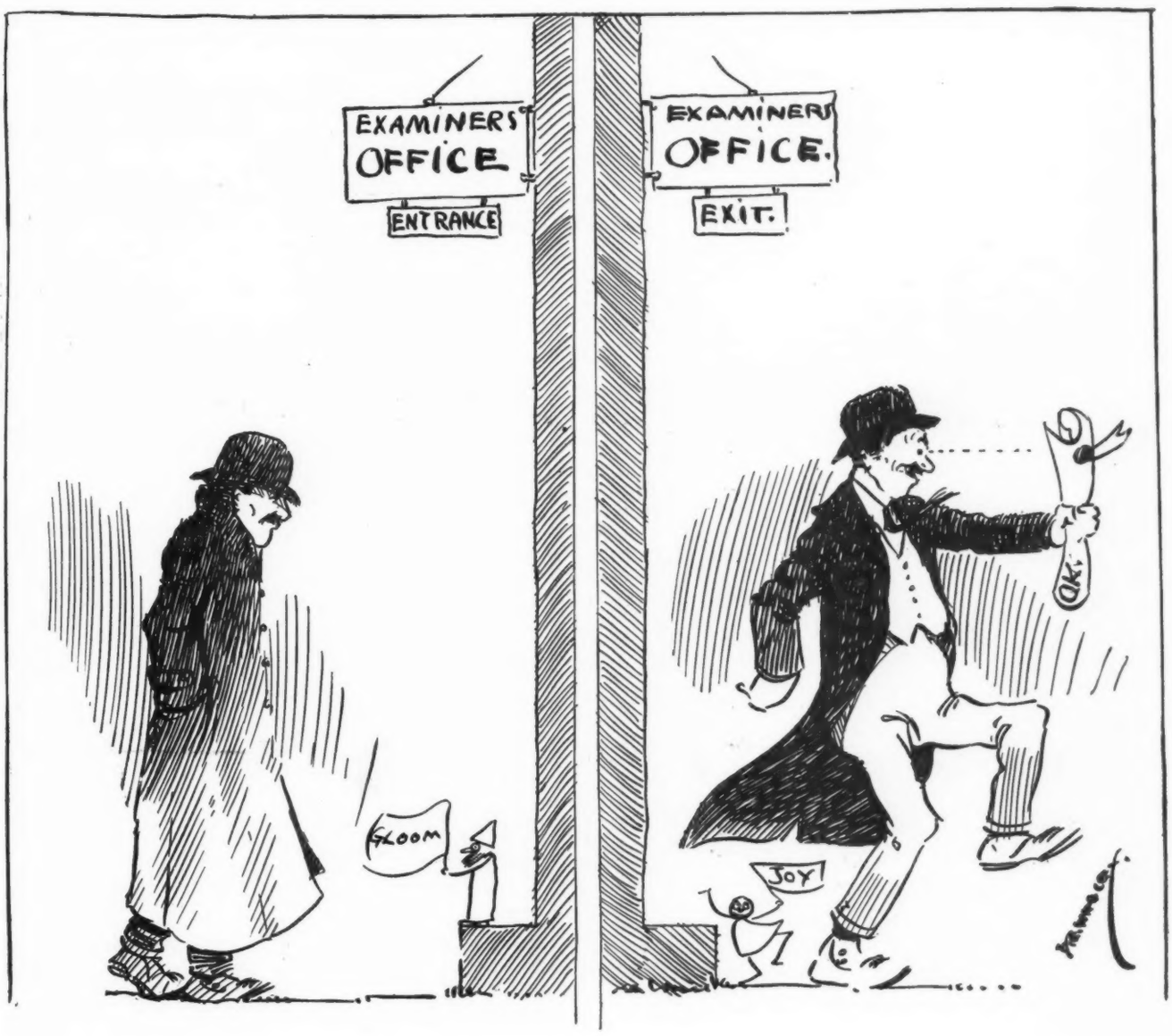


BEFORE AND AFTER TAKING

(Drawn by Delevan E. Whaley, Reading, Mich. Acknowledgment is made to T. E. Powers for the use of the manikins.)

Features of the Lemp Brewing Company's Plant

SYNOPSIS—Eight Heine-type boilers fitted with instruments for securing high economy; ancient and modern types of boiler-feed pumps and a mixed-pressure turbo-generator are the chief distinguishing features of this modern, well kept plant.

As a class, brewery power plants rank high. The machinery is almost always of good construction and well kept and a cleanly well painted appearance is characteristic. In many plants, especially in those of medium and large size, much attention is paid to efficiency, both as regards type of apparatus and methods of operation. A plant which is particularly noteworthy in these respects is that of the Wm. Lemp Brewing Co., St. Louis, Mo., which furnishes light, power, heat and refrigeration for one of the largest breweries in the world.

A general exterior view of the boiler house only is given in Fig. 1; the engine house is to the right and the piping connections between the two are all carried on the covered bridge seen extending over the railroad track.

The present boiler equipment consists of eight O'Brien-Heine type water-tube units rated at 500 hp. on a basis

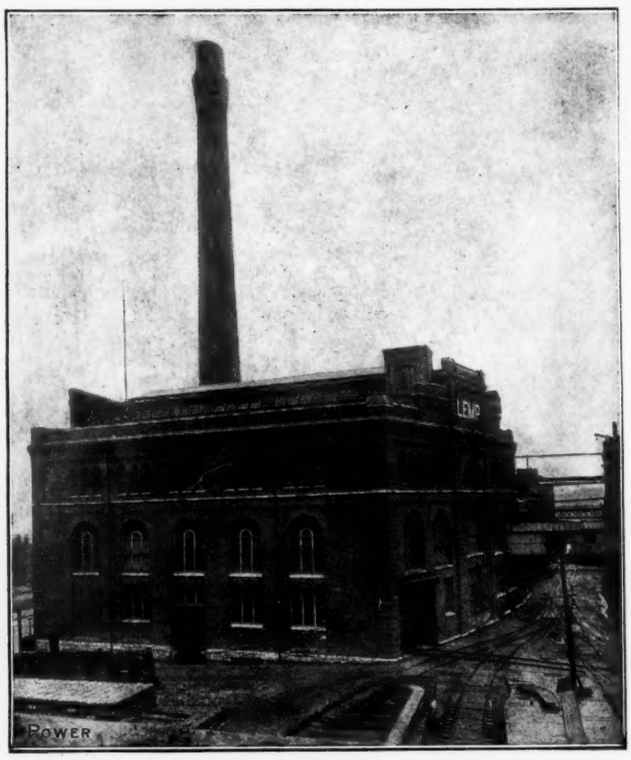


FIG. 1. GENERAL VIEW OF BOILER HOUSE

of 10 sq.ft. per hp. There is room for two more boilers of the same size. The boiler-room floor is about 15 ft. above grade level, the ash-handling floor being at grade. The working steam pressure is 125 lb., gage, per sq.in.

Coal is delivered in hopper-bottom cars which are run into the building on a trackway extending under one end. This trackway is provided with scales so that the car and shipment can be weighed just before the coal is unloaded and the weight of the empty car can be ascertained immediately after the delivery. The coal is dumped from the car directly into a crusher which discharges it to a bucket conveyor running beneath. This conveyor extends up the wall at one side, across over the bunkers and down the wall at the other side in the usual manner. The buckets are 18x24-in. in size, and the conveyor has a capacity of 80 tons per hour. The conveyor is driven by a 10-hp. motor.

ASH-HANDLING EQUIPMENT

The ash-handling arrangement is somewhat unusual. The boilers are located in two rows of four facing each other across a wide firing aisle. The ashpit hoppers are provided with special fireproof gates so arranged that the ashes can be dumped in the shortest possible time to avoid prolonged cooling of the boilers during cleaning. A narrow-gage track extends under the ash-hopper gates under each row of boilers and at each end turns and

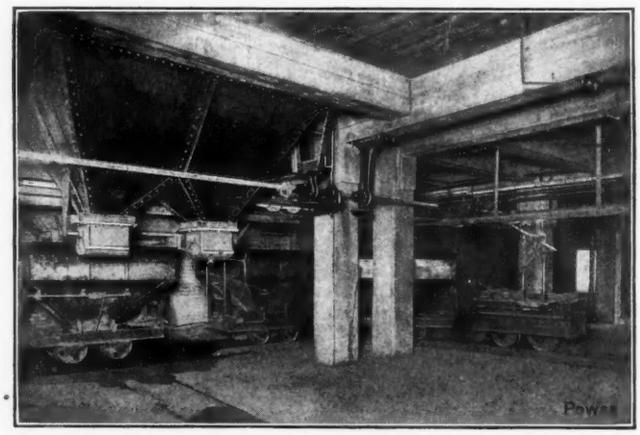


FIG. 2. VIEW IN ASH ROOM, SHOWING HOPPERS AND CARS

runs across the room to a chute opening. Small steel hopper-bottom cars, as shown in Fig. 2, placed on this track, receive the ashes from these boilers.

The cars are moved by a small electric locomotive, designed by Wm. Koedding, master mechanic for the Lemp Co. Power for this locomotive is supplied through a double-trolley pole in contact with overhead supply and return wires, suspended from a wooden strip fitted with guard strips at each side. The locomotive motor is a 2-hp. high-speed machine, speed reduction being accomplished by a worm gear which is connected with the locomotive wheels by chains and sprockets. The motor-starting mechanism is operated by a lever at either end of the locomotive. To make the locomotive run in either direction a lever is pushed in the desired direction and held in that position. The levers are counterweighted so that when the hand is removed they immediately assume a vertical position and automatically cut out the motor. Due to this fool-proof construction a special operator is not required.

The gates of the ash hoppers are of the flat, sliding type and are opened and closed by a rack and pinion. They are lined with firebrick to prevent frequently burning out. The present set of gates has been in service for about 2½ years.

The conveyor carries the ashes to the top of the boiler house and dumps them into a large storage and weighing

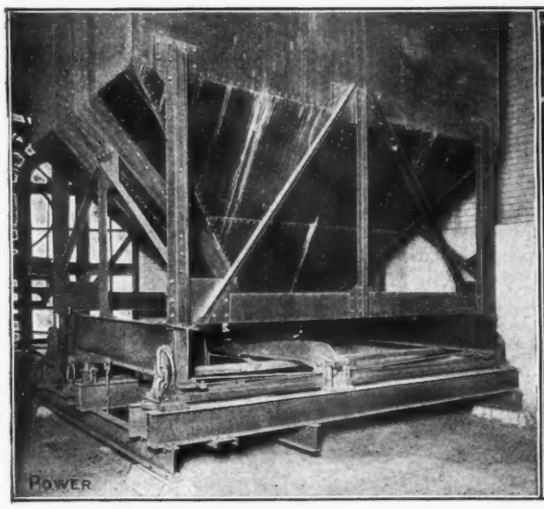


FIG. 3. ASH-STORAGE WEIGH HOPPER

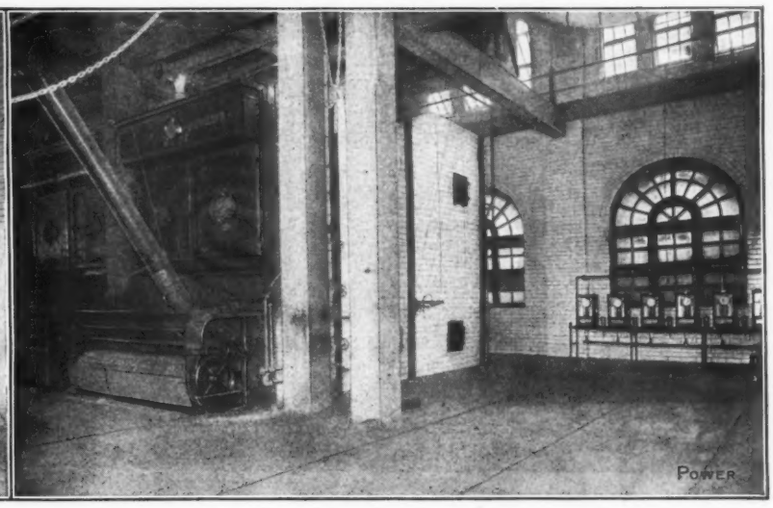


FIG. 4. VIEW IN THE BOILER ROOM

hopper, Fig. 3, which occupies a corner of the boiler room over the railroad trackway before mentioned.

The hopper is constructed of structural steel and sheet iron and lined with concrete to prevent corrosion. Its capacity is about 40 tons and it is fitted with a 40-ton beam scale. The ashes are weighed at the end of every 8-hr. shift and when an approximate carload has accumulated the total weight is taken as a check and the ashes are discharged by gravity to a car below and removed.

Siftings from the boiler stokers, which are of the chain-grate type, manufactured by the Laclede-Christie Co., are caught in bifurcated hoppers, one of which is shown in the foreground in Fig. 2. Four tracks, one under each pair of opposite boilers, extend across the ash room and each carries a car, similar to the ash cars, to receive the siftings. When a carful is obtained the car is pushed across the room by hand and the siftings are emptied into chutes which feed onto the conveyor and are carried to the bunkers above.

The stoker-driving shaft is hung on the ceiling of the ash room and the eccentric rod for each unit extends through a slot in the boiler-room floor. The stokers

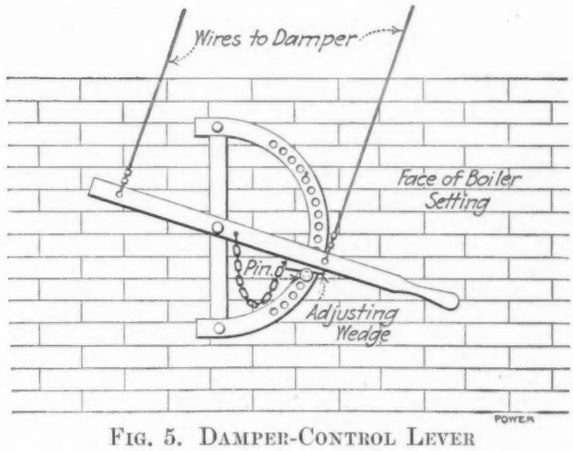


FIG. 5. DAMPER-CONTROL LEVER

are driven by a 10-hp. variable-speed motor. The speed may be changed by hand as required. A small high-speed vertical steam engine is provided for emergency use. The area of grate surface for each boiler is 100 sq.ft.

The chimney is of the radial-brick design and 250 ft. high above grates by 11-ft. inside diameter.

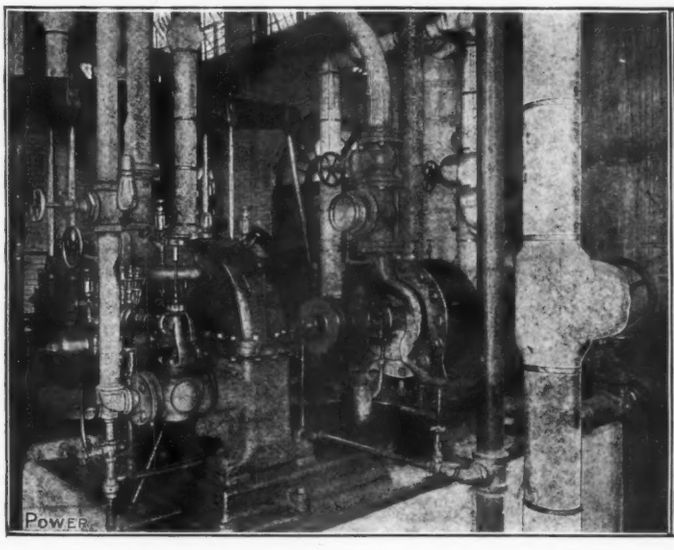


FIG. 6. MAIN BOILER-FEED PUMP AND OLD AUXILIARY

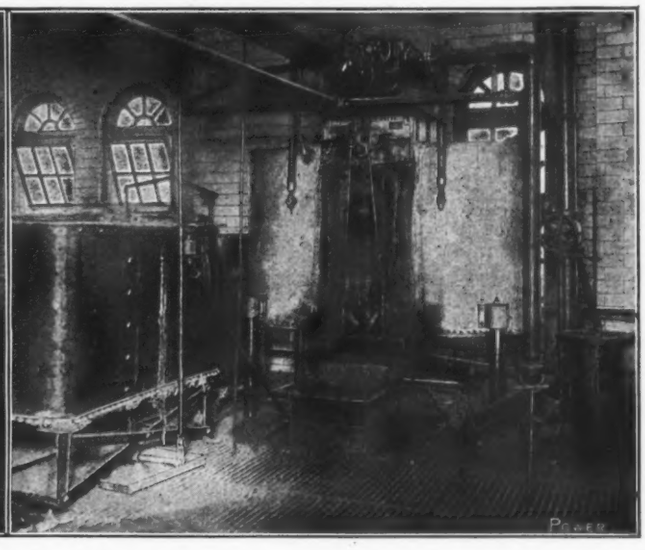


FIG. 7. WATER WEIGHER AND BOILER-COMPOUND TANK

Boiler-Room Instruments

As especial attention is given to the matter of boiler-room economy, the boilers have an unusually complete equipment of gages and instruments. A bank of six Precision Instrument Co.'s automatic CO_2 recorders are located as shown in Fig. 4, six being all that are required, as more than six boilers are never operated at any time. These instruments are piped so that any one may be connected to any boiler.

The draft is controlled by an individual butterfly damper for each boiler and adjusted by a lever and wedge, as shown in Fig. 5, located at a convenient point on the side of the setting, as may be seen in Fig. 4.

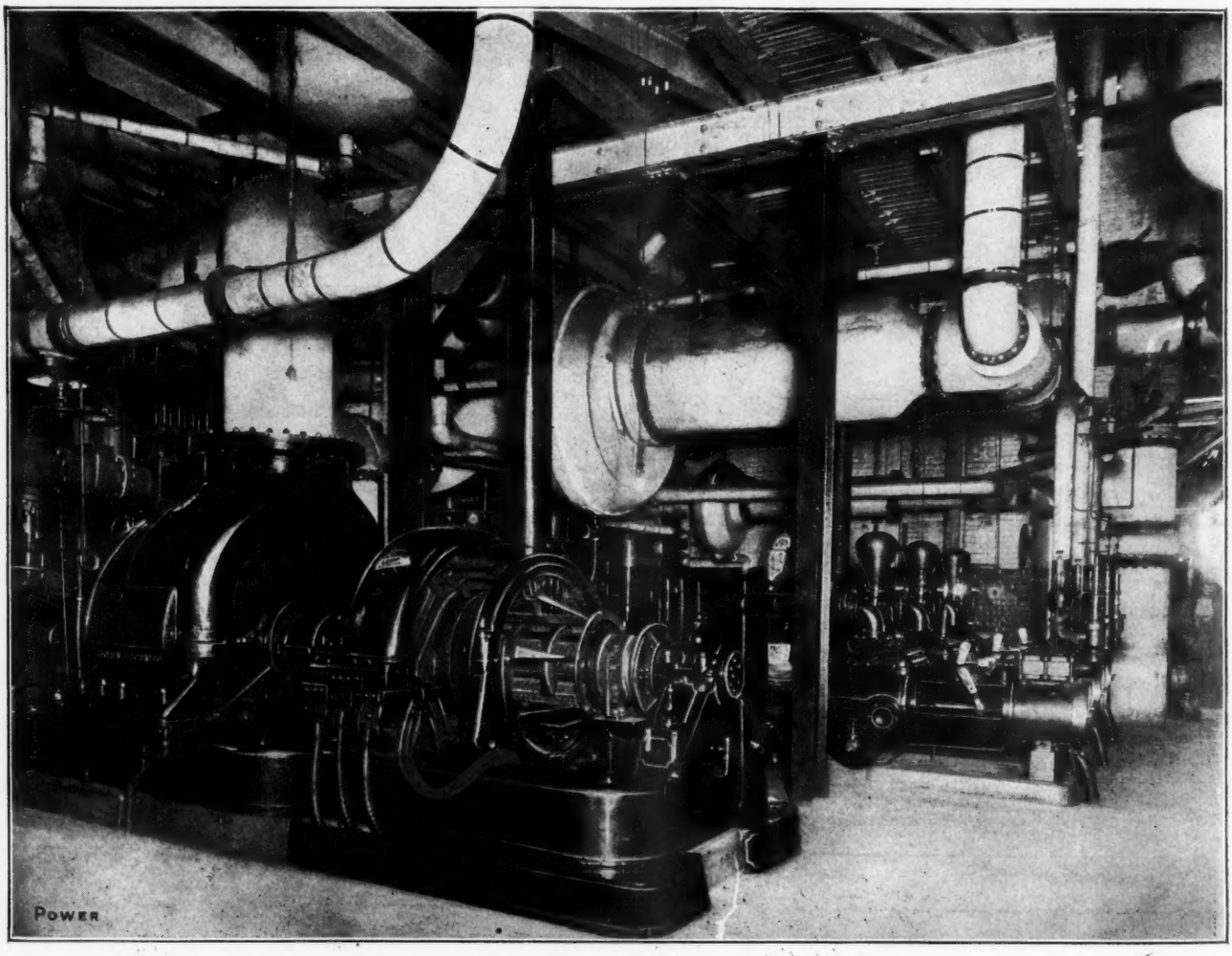


Fig. 8. Mixed-Pressure Turbo-Generator

Near the damper-control handle on each boiler setting is a differential-type draft gage which indicates the draft on the boiler side of the damper. In addition there is a recording draft-gage connected with the main uptake of each row of four boilers. A record of the steam pressure is kept by a Bristol recording-pressure gage connected with the main steam header. A recording high-range thermometer registers the flue-gas temperature at the point where the breeching enters the chimney. The flue gases are periodically analyzed by hand as a check on the automatic machines. The feed water is measured by a Worthington meter and a Hammond water weigher, one serving as a check upon the other. The boilers are equipped with Bayer soot blowers and the heat-absorbing surfaces are kept thereby in a high state of efficiency.

Arrangement is made whereby any boiler may be tested without undue inconvenience, and, consequently, frequent tests are made.

The feed water is tested regularly for scale-forming ingredients and treatment is prescribed to suit.

Boiler-Feed Equipment

The three boiler-feed pumps are an interesting study in the art of steam-pump development. First, there is an ancient walking-beam steamboat type which has seen over 25 years of service. The beam, some of the water cylinders and connecting rods, and part of the crank disk, may be seen in Fig. 6. There are four 8x16-in. plunger-type water cylinders situated next to the center column supporting the walking-beam. Beyond these cylinders at the left is the one steam cylinder, which is 10x32 in. in size and double acting. Beyond the water cylinders at the right are the crank and flywheel.

This pump, although capable of good economy, as boiler-feed pump economies go, is now a sort of pensioner, and only used in cases of extreme emergency. Alongside of it and occupying the foreground of Fig. 6 is a turbine-driven centrifugal pump, typifying the latest practice in boiler-feed apparatus. This pump, with a capacity of 350 gal. per min., is the main boiler feeder. The third pump, an Epping-Carpenter 12x8x12-in. hori-

zontal duplex, located to the right of the turbine-driven unit, is employed only for emergency service.

The feed water is pumped from a closed heater in the engine house to an open Cochrane heater located at a high point in the boiler house, whence it flows by gravity to the water weigher, which is on a level with the top of the boilers. From this weigher the water is discharged by gravity to an 8x20-ft. coagulating tank and a filter below on the boiler-room floor, from which the feed pump receives its supply. The feed water goes to the boilers at a temperature of from 190 to 200 deg. F.

At each discharge of the weigher a small pump on the boiler-compound tank which is shown at the left in Fig. 7 is operated by means of an arrangement of levers and a definite quantity of the compound is injected into the water while the latter is being discharged.

Water is purchased from the city and after passing through the city meters is distributed by a 10-in. Worthington centrifugal pump, driven by a 165-hp. Curtis horizontal single-stage non-condensing turbine at a speed of 2000 r.p.m. The pump-discharge pressure varies from 80 to 100 lb.

Mixed-Pressure Turbine

Refrigeration is provided by five De La Vergne double-acting compressors, three rated at 170 and two at 220 tons. All are driven by horizontal Corliss engines. The exhaust from the 36x36-in. steam ends of the two larger machines is used in a Curtis mixed-pressure turbine to drive a 500-kw. direct-current generator, Fig. 8. If needed, the exhaust from the 32x36-in. steam ends of the three smaller machines may also be supplied to the turbine through a bypass. Ordinarily, however, these operate condensing. In order to be able to change from condensing to non-condensing operation easily and quickly, Mr. Koedding has fitted an adjusting mechanism into the rods connecting the exhaust valves and wristplate on all the machines so that the length of these rods may be changed without shutting down.

This turbo-generator operates at a speed of 1500 r.p.m. and produces direct current for three-wire distribution at 250 and 125 volts. Its installation has eliminated the use of three 200-kw. vertical-compound condensing Corliss engines and effected a marked economy in the overall steam consumption of the plant.

In addition to the rather complete equipment of instruments for securing boiler economy the plant is fitted with General Electric steam-flow meters for measuring the steam supplied to the various departments. There is also a portable recording-flow meter for testing and checking the demands of the different departments, and a portable indicating-flow meter for testing any piece of steam-consuming apparatus when desired.

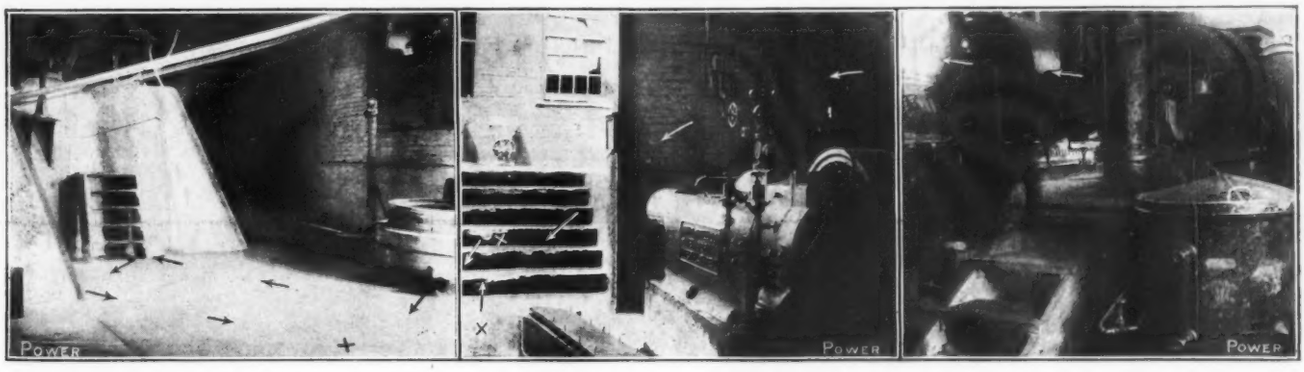
Any changes made in the design or equipment of the plant are decided upon and worked out by an engineering board consisting of Wm. J. Lemp, president of the company, who is a mechanical engineer, J. T. Norton, architect, and Wm. Koedding, master mechanic. The operation of the plant is under the direct supervision of the last named.

Turbine Rotor Climbs Stairs

It is not every turbine rotor that performs the phenomenal feat of bursting through its casing, jumping over an air compressor and climbing stairs leading into the yard, all in one nonstop dash. Yet a 100-kw. De Laval impulse turbine in the plant of the Alpha Portland Cement Co., Phillipsburg, N. J., claims this distinction.

Determined to preserve the reputation it thus established it endeavored to try it all over again about two weeks after the first "break."

This is how it happened: The machine, whose normal speed was 9000 r.p.m., drove two small generators through gears. Due to some cause, not explained, the speed be-



Arrows Indicate the Course Taken by the Rotor of the Turbine at the Right

The exhaust-steam supply line to the turbine is equipped with a large Cochrane oil-and-water separator having a capacity of 50,000 lb. per hr. Normally, the steam enters the turbine at a vacuum of about 3 in. of mercury and discharges into a vacuum of 27 or 28 in. developed by two Alberger surface condensers of 3000 sq.ft. each. The condensed steam is used in a 250-ton can ice plant. The live-steam supply to the turbine is cut in automatically when the available exhaust steam becomes insufficient.

came excessive. The rotor burst through the casing, knocked away a 6-in. exhaust pipe, leaped over a small air compressor onto the floor in front of the cement stairs leading out of the engine room. From here it darted for the stairs, successfully climbing them until it reached the third step, where, owing to its excessive speed, it lost its toe hold and ground out about 5 in. of the step before turning back into the room previous to a 35-ft. lunge into an inoffensive foundation. Wheeling quickly to the home stretch it crashed into another similar foun-

dation 10 ft. away and then sped across the floor nearly to the starting point.

The arrows in the photographs show the route taken by the rotor. When the machine received the necessary repairs, and about two weeks after this startling piece of misbehavior, it again sought liberty by endeavoring to do the stunt over again and would have succeeded but for the presence of the engineer who foiled it by shutting off the steam.

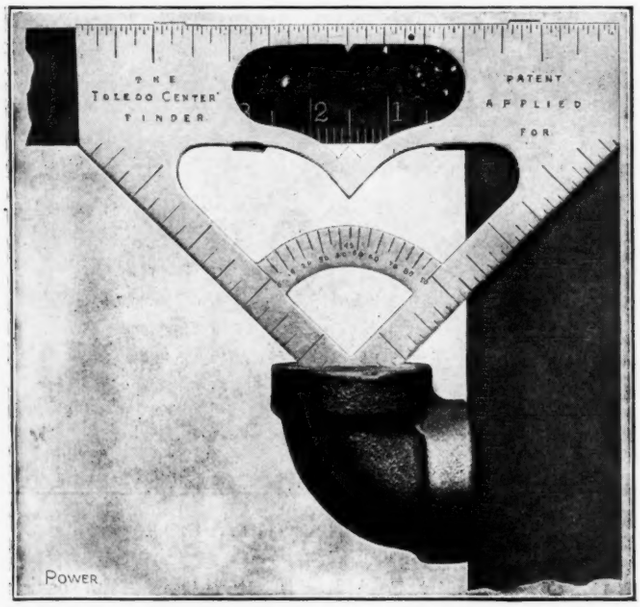
The second accident was caused by a defective gear shaft, which broke. The effect was like an increase of load and the governor on the opposite shaft, operated to admit more steam. Before the speed became damaging the engineer shut off the steam.

Examination of the shaft showed that it was hollow for at least 9 in. of its length. The break occurred 4 in. from the flange on the generator end. The cracks or hollow extending into the shaft about 2¼ in. The cracks in the other end, which was bent nearly U-shape, were much deeper, being about 6¾ in.

The cross-sectional area of the shaft was reduced so much as to allow for no undue strains, and it would appear that a sudden increase of load snapped the shaft. Fortunately, no one was injured in either accident.

Toledo Center Finder

This tool is made to use with a carpenter's steel square, as illustrated, when it will readily determine the center measurement of pipe fittings; it covers a range of sizes from 1 to 6 in. It is made of steel, with a ground face. Each straight-edge is graduated, one edge is $\frac{1}{16}$ in., one



TOLEDO CENTER FINDER

⅛ in., and one is ¼ in. The cross brace of the finder forms an arc of a circle and serves as a protractor, being graduated down to 5 deg.

In addition to its use as a center finder, the instrument can be used for striking correct angles and inclinations in laying out piping.

The method employed to find the center of a fitting is to hold the same true against the wide tine of the square and push up against the finder, which will slide one way or the other on the square until it seats itself in the fitting. The pointer on the finder will then show the measurement desired. The finder is made by the Toledo Pipe Threading Machine Co., Toledo, Ohio.

Improved Falls Engine Stop

The improvement recently made to this engine stop is confined to the arrangement of the front-head mechanism, whereby the solenoid is mounted in the upper half

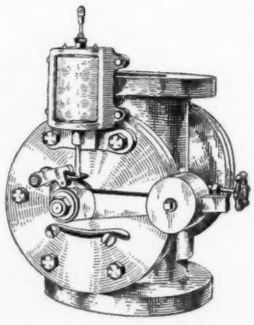


FIG. 1. SOLENOID ON VALVE

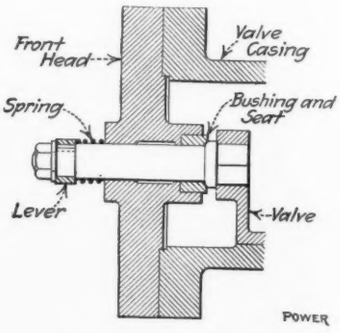


FIG. 2. PACKINGLESS STUFFING-BOX

of the valve, Fig. 1, and to the simplified speed-limit band in which the pivoted bearing has been done away with. An improvement has also been made by the manufacturer, the Falls Machine Co., Sheboygan Falls, Wis.,

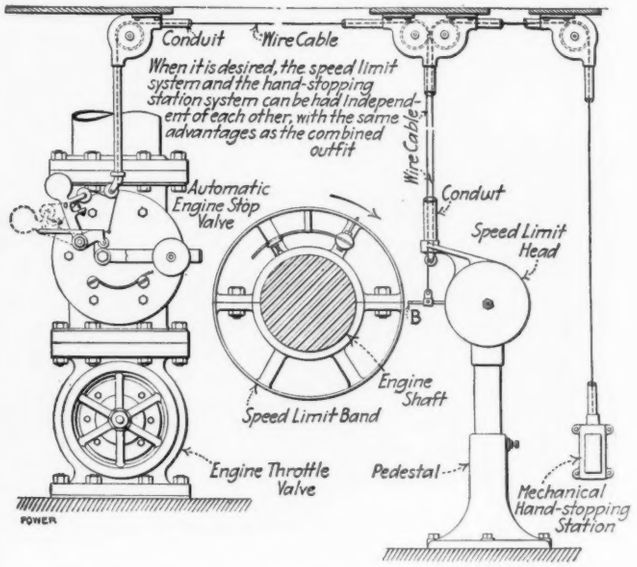


FIG. 3. STOP VALVE, SPEED-LIMIT BAND AND CONNECTIONS

in the elimination of the stuffing-box in the front head of the valve, as the joint has been made packingless, Fig. 2. The steam pressure inside the valve-stem shoulder acts against the squared end of the valve stem and holds it firmly seated against the valve-stem bushing and seat. The contact face on the valve-stem shoulder is curvilinear, its corresponding seat in the valve-stem bush-

ing being of a straight bevel form. The two faces are lapped together to form a ground joint, which eliminates leakage of steam.

Referring to Fig. 3, which shows the speed-limit band, should the cable break at any point the weight lever on the front head is released, which action occurs if the stop is operated from a hand stopping station or by the speed-limit band.

Pacific Mills Coal- and Ash-Handling Equipment

SYNOPSIS—A coal- and ash-handling system, consisting of three distinct parts. The first is a pivoted bucket conveyor to transfer the coal from the cars to the coal bunkers. The second part consists of charging cars for conveying coal from the bunkers to the boiler room, and the third part is an industrial railway system for removing ashes.

The completeness of the coal- and ash-handling apparatus in the power plant of the new print works of the Pacific Mills, South Lawrence, Mass., shows that large industrial plants appreciate that economy can be obtained by employing uptodate apparatus in the power plant as well as by the use of improved machinery in the manufacturing departments.

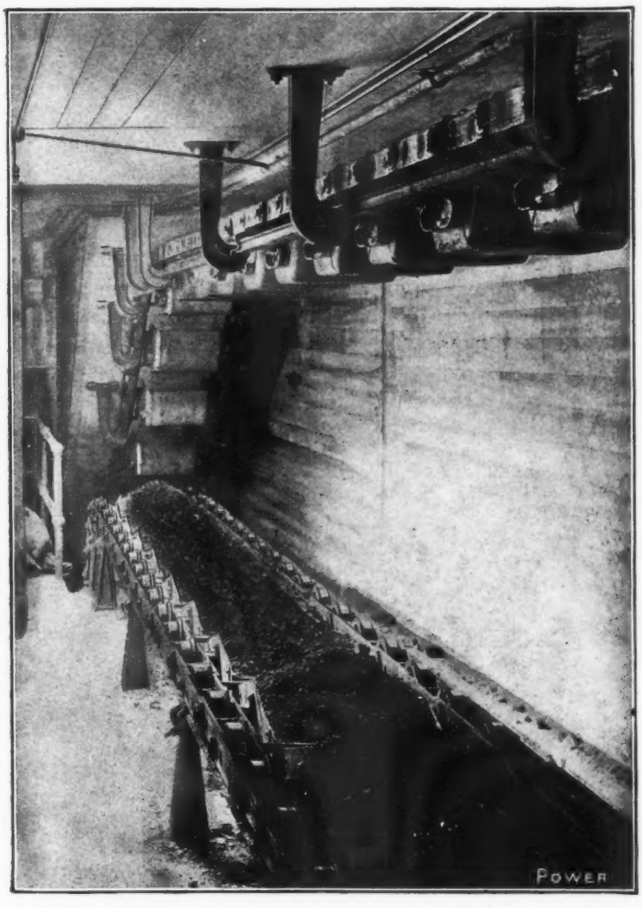


FIG. 1. PIVOTED BUCKET CONVEYOR

There are three distinct parts to this equipment. The first consists of a pivoted-bucket conveyor which receives the coal from the railroad cars and distributes it into the storage pocket. In addition to this conveyor a motor-driven car distributes coal to parts of the storage pocket not reached by the conveyor. The second part, consisting of charging cars and an industrial railway, is used to convey coal from the storage pocket to the boiler room. The third part, consisting of another industrial railway system, is for the removal of ashes.

Coal is brought to the plant in railroad cars of the bottom-dumping type and is distributed into a pit under the track. It then passes into an electrically driven coal cracker, which breaks up the large chunks of bituminous coal into lumps of proper size for firing. The coal cracker discharges through a revolving filler to a pivoted-bucket conveyor which runs through the tunnel, shown in Fig. 1, and then rises, as shown in the rear of Fig. 2, and carries coal to the distributing chutes, each of which is provided with an automatic trip so that the coal can be discharged from the conveyor buckets into any chute desired.

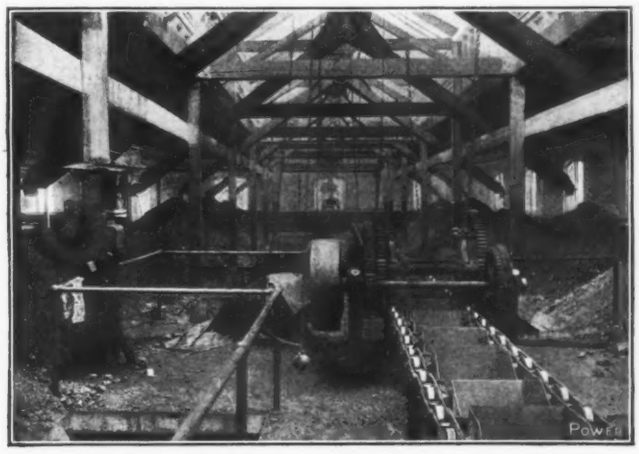


FIG. 2. TOP OF COAL BIN, VERTICAL-RUN CONVEYOR AND EMPTY RETURN BUCKETS

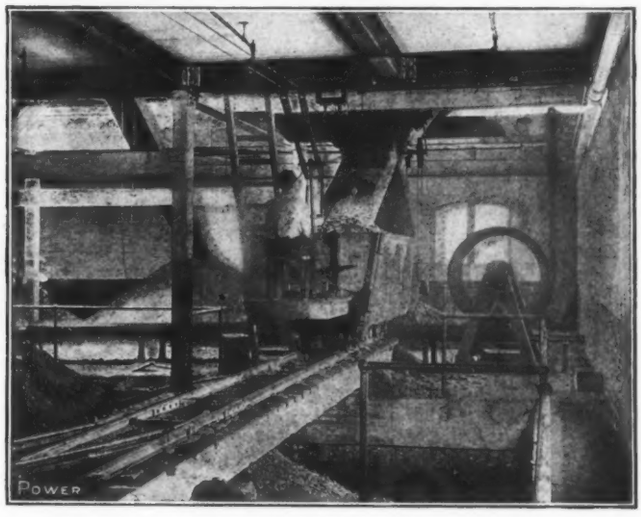


FIG. 3. ELECTRICALLY DRIVEN DUMP CAR

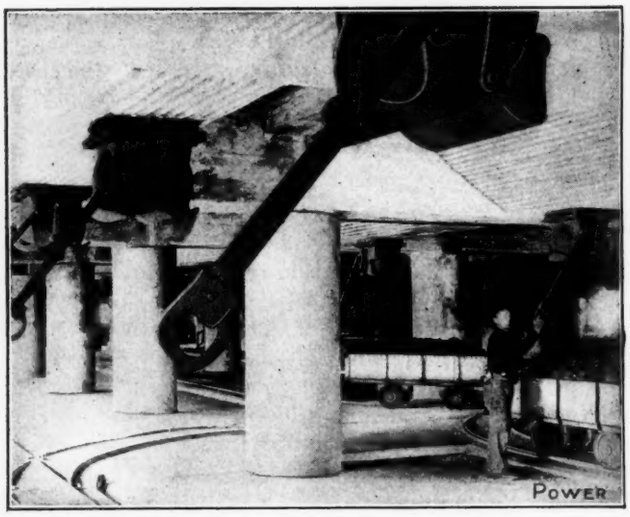


FIG. 4. BOILER-ROOM CHARGING CAR AND LOADING HOPPERS

The motor-driven conveyor driver is located on the upper level. The conveyor is driven by an open-geared pawl driver.

Besides the coal pocket shown in Fig. 2, there is an auxiliary row of storage bins at the extreme left of the building. For filling these bins, the conveyor can be arranged to discharge into an auxiliary hopper, Fig. 3, which is used for filling an electrically operated car of 2-ton capacity, running on a track over the storage bins.

The bins have a combined capacity of 10,000 tons of coal, but are so subdivided that the individual bins can

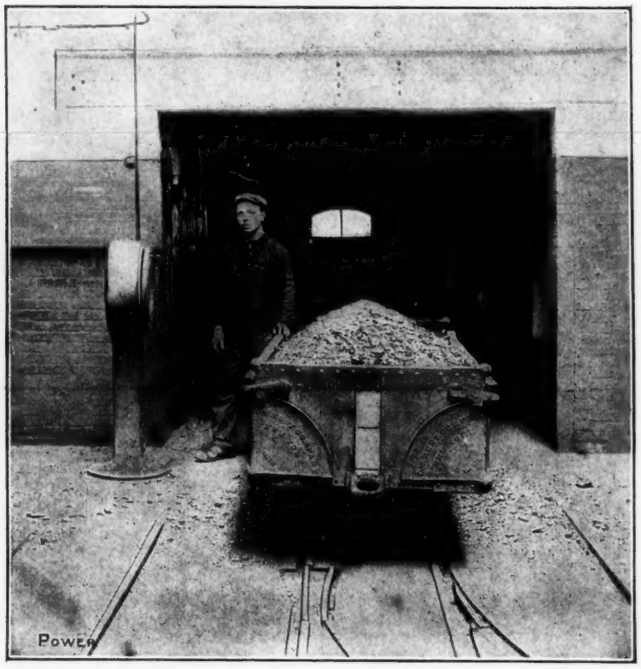


FIG. 5. WEIGHING THE COAL

be emptied in a few minutes in case the coal commences to heat and there is danger of spontaneous combustion.

Taking up the second part of the system; from the bins shown in Fig. 2, the coal is drawn off through duplex valves into boiler-room charging cars, as shown in Fig. 4. Although these cars are of 1-ton capacity, they can be filled in a very short time—less than a minute.

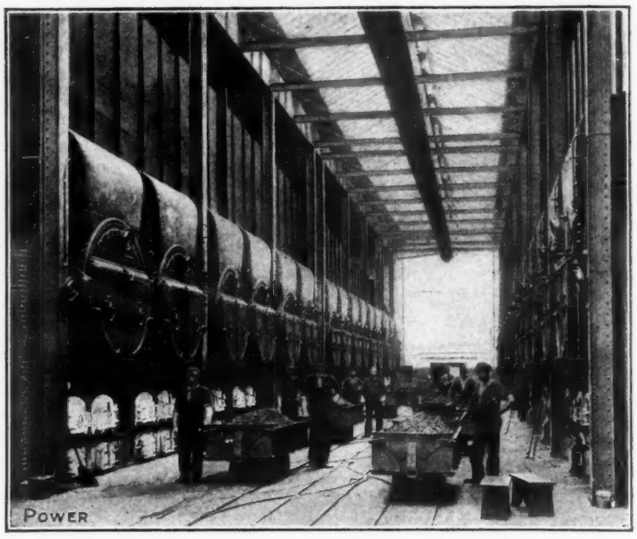


FIG. 6. CHARGING CARS IN FRONT OF BOILERS

The charging cars are pushed by hand onto the scales, Fig. 5, and a record of all the coal burned is kept. The cars are then pushed into the boiler room, as shown in Fig. 6. The two outside tracks are so located that the fireman merely drops the side of the car and shovels the coal into the furnace. The middle track is provided for shunting out empties and running in full cars.

These coal cars are designed especially to facilitate firing the coal into boiler furnaces. They are made of sheet steel, stiffened with angle iron, the corners rounded off for the workmen's hands and the bottom flush riveted. When the side door is let down, its bottom is slightly higher than the floor of the car, so that the fireman will not catch his shovel when scooping up the coal.

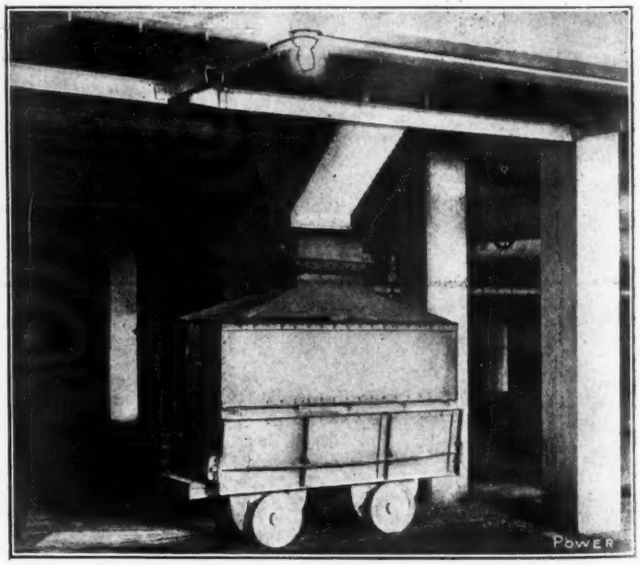


FIG. 7. ASH CAR AND ASH CHUTE

This completes the coal-handling equipment which has a capacity of 60 to 70 tons an hour, more than ample to keep the 30 boilers supplied when running at full capacity.

Fig. 7 shows the room underneath the boilers, for the removal of ashes. The ash car has been designed so as to be practically dust-proof. As soon as the car is run

under the ash hopper, a lever at the top of the car is pulled down and a telescoping spout on the car is shoved up and encircles the bottom of the ash hopper, thus preventing ashes from flying around.

This coal- and ash-handling system was designed by Lockwood, Greene & Co.; the apparatus, including the conveyors, charging and ash cars, motor-driven coal car, coal and ash valves, trackage, etc., used in this equipment, was supplied by the C. W. Hunt Co., Inc., West New Brighton, N. Y. The cars have outside-flanged wheels, permitting one man to easily push a 1-ton load around a curve of 12 ft. radius, as in taking the curves the outside wheels run on their flanges so that the tread of the inner wheel and the flange of the outer one fall upon the theoretical cone of revolution.

Maihak Indicator and Boettcher Efficiency Indicator

The Maihak indicator, Fig. 1, is designed to be used especially with gas and Diesel oil engines. It is reliable up to 600 r.p.m. and is fitted with a piston one-fifth the area of the normal piston.

The spring support through which the piston rod passes centrally forms the cylinder cover and also the turntable for the pencil link work.

The indicator is convertible from a right-handed to a left-handed instrument. The link attachment, piston and spring may be instantly removed. The piston rod is hollow and removable. The pencil lever has a magnifying ratio of 6 to 1 and the link gives a practical straight line motion. The indicating link work is a modification of the pantograph.

A double spring is used and is arranged to operate in tension, instead of compression. The two spirals forming the spring are screwed into a base formed in two parts and rigidly fastened.

The drum is controlled by a spiral spring and is fitted with a double cord whereby two indicators may be coupled together by a common reducing link work. It is fitted with a friction clutch to facilitate connecting the cord to the reducing motion.

The indicator may be fitted with a Böttcher's efficiency reckoner, by means of which the mean indicated efficiency of the engine may be found by reading a counting instrument and multiplying by a constant. The reckoner takes account of every stroke of the engine and counts every diagram taken; the indicator can be used with or without the instrument.

Referring to Figs. 2 and 3 the reckoner is supported on a pillar *A*. The counting roller *E*, in the frame *D*, on the upper front surface of the indicator drum, is pushed to and fro in a radial direction on top of the paper drum by a lever *C*. A spring *F* presses the roller against the upper surface with a known pressure of adhesion. Upon the roller spindle a worm gear is mounted which transmits the motion to a simple set of spur wheels capable of registering a number of observations extending over a long period.

The travel of the periphery of the counterspool is directly proportioned to the average pressure indicated, and this records the power expended. To find the mean pressure exerted during a definite period of time, a revolution counter must be read in addition to the pressure counter, but if the average indicated horsepower is to be ascertained it is not necessary to use a revolution counter.

These instruments are put on the market in this country by Herman Bacharach, 1009 Hartje Bldg., 14 Wood St., Pittsburgh, Penn.

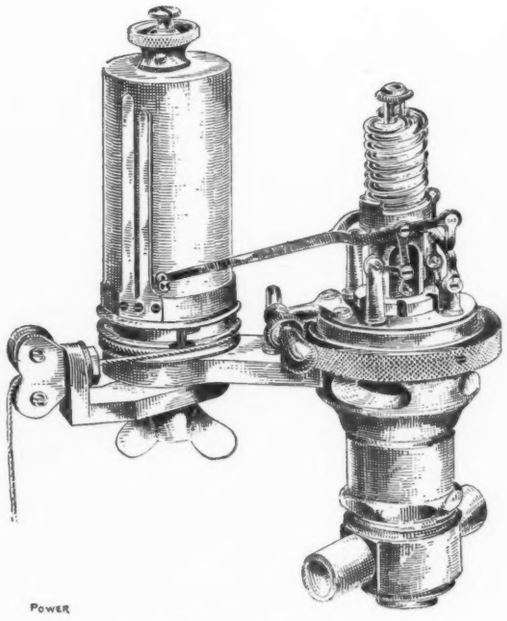


FIG. 1. MAIHAK INDICATOR FOR GAS AND OIL ENGINES

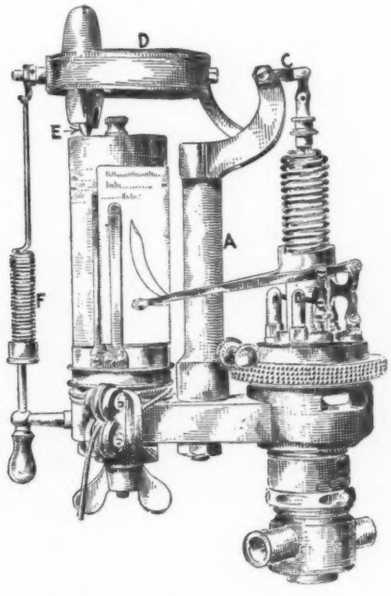


FIG. 2. COUNTER IN OPERATIVE POSITION

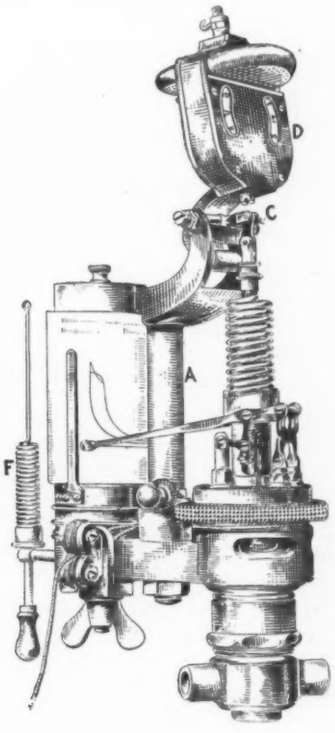


FIG. 3. COUNTER IN LIFTED POSITION

CO_2 Thermoscope

An instrument for ascertaining the quantity of CO_2 in the waste gases from a boiler furnace is shown in Fig. 1. The CO_2 thermoscope makes use of a well known principle of thermo chemistry, but one which has not heretofore been applied in the analysis of gases, viz., that chemical reactions are attended by change of temperature, and that the amount of heat either absorbed or disengaged is determined by the nature of the reaction, and the quantities of reagents which enter into combination.

When carbonic-acid gas (CO_2) either pure or in admixture with air is brought into contact with caustic soda a chemical reaction takes place, and heat is evolved, the amount of which is proportional to the quantity of CO_2.

If, therefore, a measured quantity of the gas mixture to be analyzed is brought into contact with caustic soda, the amount of which being somewhat more than sufficient to absorb all the CO_2 present, it is only necessary to measure the heat evolved to obtain a measure of the CO_2 in the mixture.

The caustic soda is prepared for use in hermetically sealed metallic cartridges, a fresh cartridge being em-

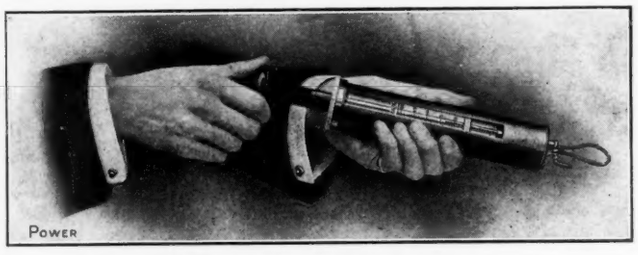


FIG. 1. CO_2 THERMOSCOPE

ployed for each analysis. The heat reaction is produced by passing a known quantity of the gas mixture to be analyzed through the contents of the cartridge, while the amount of heat disengaged is measured by means of a specially constructed calorimeter thermometer using a scale empirically calibrated in percentages of CO_2.

With the exception of two thermometers the instrument is made of brass and hardwood. It is strongly made and will stand hard usage. It is 10 in. in length and 2 in. in diameter and may be carried in the pocket. Thus an engineer with a few cartridges in one pocket and the CO_2 thermoscope in the other, and without the necessity of setting up and adjusting any apparatus can obtain analysis of the CO_2 from a range of boilers quickly.

After it has been used the instrument requires no washing or cleaning. It is just put back into its leather case and is ready again for instant use the next day or the next month as the case may be.

The instrument consists of three essential parts. A cylinder is fitted with a plunger for drawing a measured quantity of the mixture from the flue or chamber containing the gas mixture, and subsequently passing it through a cartridge-shaped receptacle containing pulverized caustic soda in which the heat reaction occurs. There is also a thermometer with its bulb constructed to surround or jacket the cartridge, so that the heat of reaction can be imparted to the mercury, the amount of its expansion, i.e., the percentage of CO_2 to be observed on a movable scale.

By reference to Fig. 2 it will be seen that *B* is a cylinder fitted with a piston and cup leathers. The thermometer *E* is conveniently mounted on the cylinder, and the whole is inclosed by the outer cylindrical metallic jacket *A*, which is slotted to show the stem of the thermometer. A movable scale *F* calibrated in percentages of CO_2, is arranged to slide in the slot. *D* is a cartridge of thin sheet metal containing the pulverized caustic soda, and

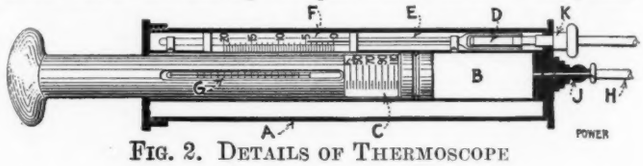


FIG. 2. DETAILS OF THERMOSCOPE

is shown jacketed by the thermometer bulb in position for analysis.

Connection is made between the cylinder and cartridge by means of the rubber tube *K* and *H*, the latter being attached to the tap *J*.

In order to correct the volume of the gas taken into the cylinder for varying room temperatures, the length of travel of the piston is regulated according to the temperature, as shown on the thermometer *G* by the scale *C* engraved on the piston rod.

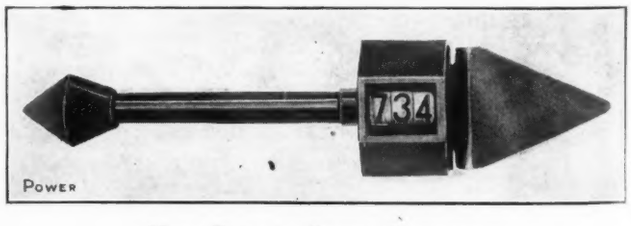
When about to make an analysis a cartridge is pricked at each end to enable the gas to flow through it. It is then attached to the rubber connection and inserted in the thermometer bulb. A sample of gas is drawn into the cylinder, the necessary temperature correction being made, and the free end of the rubber tube is connected to the cylinder nozzle. The zero of the movable scale is adjusted opposite the top of the mercury column of the thermometer and the piston rod is depressed. The gas mixture flows through the cartridge, heat is generated, the mercury rises, and opposite the highest point reached by the rising column may be read from the scale in plain figures the percentage of CO_2. The reading of the instrument may be taken in any position.

The CO_2 thermoscope is manufactured by the Underfeed Stoker Co., Ltd., of Coventry House, South Place, London.

❋

Vest-Pocket Shaft-Speed Counter

A neat little speed counter, suitable for carrying in a vest pocket, is illustrated herewith. The counter reads right or left. One end is made with a metal arrow point for cleaning out the counter-sunk end of a shaft of any accumulation of dirt or oil. The end of the counter spindle is applied to the center of the shaft.



VEST-POCKET SPEED COUNTER

It is not necessary to start the readings with zero, as it is only necessary to deduct the initial reading from the final result, which will give the number of revolutions per minute. The counter is made by the American Steam Gauge and Valve Manufacturing Co., Camden St., Boston, Mass., and sold at a moderate price.

ELECTRICAL DEPARTMENT

Transformer Characteristics

By Edward T. Moore*

A transformer should be rated by its kilovolt-ampere (kv.-a.) output. If it is supplying a load having a power factor of 100 per cent., the kilowatt output will be the same as the kilovolt-ampere output. If, however, it is supplying a load having a power factor less than 100 per cent., the kilowatt output will be less than the kilovolt-ampere output in the ratio that the power factor is less than 100 per cent. For example, a transformer having a full-load rating of 100 kv.-a. will have a full-load rating of 100 kw. at 100 per cent. power factor, 90 kw. at 90 per cent. power factor, 80 kw. at 80 per cent. power factor, etc.

Ratings for Special Groupings

For certain groupings of single-phase transformers on polyphase circuits, the rating of the group is less than the sum of the single-phase ratings of the individual transformers composing the group. This point can be readily understood by reference to the following illustrations; the first two cases of delta and star connections, being very familiar, are given merely as explanatory steps to the other cases. For a three-phase circuit, assume the following general conditions:

Line voltage $= E$.

Line current $= \sqrt{3}\, I = 1.73\, I$.

Total kv.-a. to be transformed $= 1.73\, IE\, \sqrt{3} = 3\, IE$.

1 to 1 ratio of transformation.

1. Three transformers delta-connected, assuming each transformer to have a capacity $= \frac{3\, EI}{3} = EI$ kilovolt-amperes.

Transformer voltage = line voltage $= E$. (See Fig. 1).

Transformer current = line current $\div 1.73 = I$.

Kilovolt-amperes in primary winding of each transformer EI.

The kilovolt-amperes in the secondary winding are the same. Therefore, the assumed single-phase rating that each single-phase transformer equals one-third the group rating, is correct.

2. Three transformers, star-connected, assuming each transformer to have a capacity of $\frac{3\, EI}{3} = EI$ kilovolt-amperes.

Transformer voltage = line voltage $\div 1.73 = \frac{E}{1.73}$ (See Fig. 2).

Transformer current = line current $= 1.73\, I$.

Kilovolt-amperes in primary winding of each transformer $= \frac{E}{1.73} \times 1.73\, I = EI$.

The kilovolt-amperes in the secondary winding are the same. Therefore, the assumed single-phase rating that each single-phase transformer equals one-third the group rating is correct.

3. Two transformers, V-connected, or open delta, for three-phase operation, assuming each transformer to have a capacity of $\frac{3\, EI}{2} = 1.5\, EI$.

Transformer voltage = line voltage $= E$. (See Fig. 3).

Transformer circuit = line current $= 1.73\, I$.

The kilovolt-amperes in the primary winding of each transformer $= 1.73\, EI$.

The kilovolt-amperes in the secondary winding are the same. Therefore, each transformer must carry $1.73\, EI$ kilovolt-amperes in order to deliver $1.5\, EI$ kilovolt-amperes to the secondary circuit. Therefore, the single-phase rating of each single-phase transformer must be 1.155 times one-half the group rating.

4. Two transformers, T-connected, assuming each transformer to have a capacity of $\frac{3\, EI}{2} = 1.5\, EI$ kilovolt-amperes.

Main transformer voltage = line voltage. (See Fig. 4).

Main transformer current = line current $= 1.73\, I$.

The kilovolt-amperes in the primary winding $= 1.73\, EI$.

The kilovolt-amperes in the secondary winding are the same.

"Teaser" transformer voltage $= 0.866 \times$ line voltage $= 0.866\, E$.

"Teaser" transformer current = line current $= 1.73\, I$.

The kilovolt-amperes in the primary winding of teaser $= 1.73\, I \times 0.866\, E = 1.5\, EI$.

The kilovolt-amperes in the secondary winding are the same.

Although the kilovolt-amperes in the "teaser" transformer windings are $1.5\, EI$, the excess current in the windings is the same as in the main transformer windings, as duplicate transformers are used and the currents in each are equal. This shows that the main transformer must carry $1.73\, EI$ kilovolt-amperes to deliver $1.5\, EI$ kilovolt-amperes and the "teaser" transformer $1.5\, EI$ kilovolt-amperes, but with excess current. Therefore, the single-phase ratings of each single-phase transformer must be 1.155 times one-half the group rating.

5. Two transformers connected for three-phase to two-phase operation, assuming each transformer to have a capacity $= \frac{3\, EI}{2} = 1.5\, EI$.

Main transformer voltage = line voltage $= E$. (See Fig. 5).

Main transformer current = line current $= 1.73\, I$.

Kilovolt-amperes in primary of main transformer winding $= 1.73\, EI$.

*Electrical engineer, Westinghouse Electric & Manufacturing Co.

"Teaser" transformer voltage = 0.866 × line voltage = 0.866 E.

"Teaser" transformer current = line current = 1.73 I.

The kilovolt-amperes on the primary side of the "teaser" transformer is 1.73 × 0.866 E = 1.5 EI, but there is an excess current in the windings equal to that in the main transformer, as the transformers are duplicates and the currents in each are equal. The main transformer, therefore, carries 1.73 EI kilovolt-amperes in the primary or three-phase winding to deliver 1.5 EI kilovolt-amperes, and the "teaser" transformer 1.5 EI kilovolt-amperes, but with excess current (1.73 I × 0.866 E = 1.5 EI). Therefore, the single-phase rating of each single-phase transformer must be 1.155 times one-half the group rating if based on the average of the current in both the three-phase and two-phase windings.

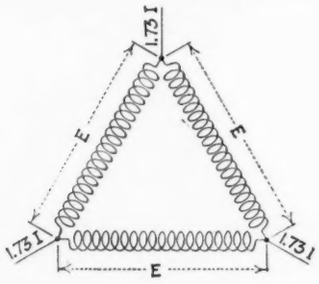


FIG. 1

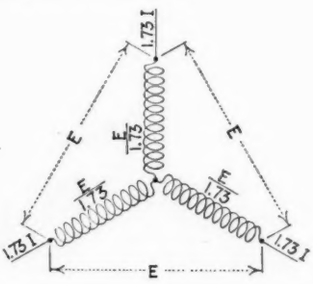


FIG. 2

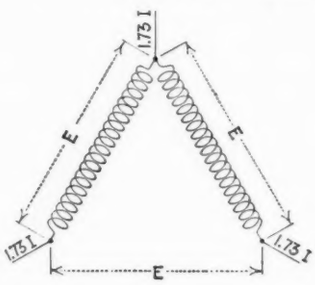


FIG. 3

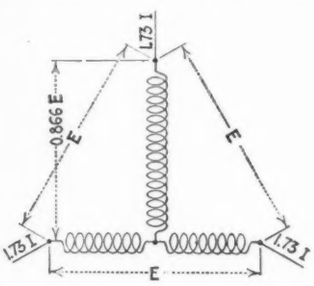


FIG. 4

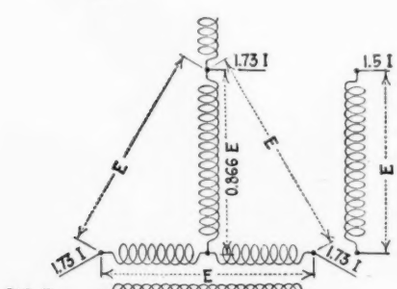


FIG. 5

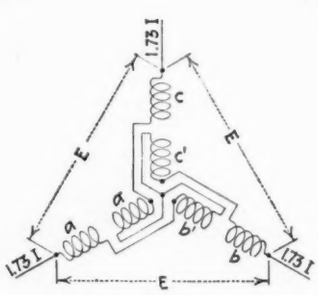


FIG. 6

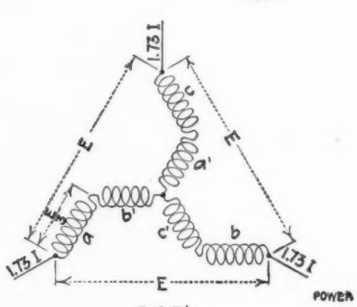


FIG. 7

DIAGRAMS OF DIFFERENT TRANSFORMER CONNECTIONS

6. Three transformers with the secondary windings connected in interconnected star for operating a three-wire, direct-current, three-phase rotary converter assuming each transformer to have a capacity of $\frac{3\,EI}{3} = EI$.

The primary side of the transformer will be delta or star connected and the conditions will be normal, as given in cases 1 and 2. The secondary will be connected as shown in Fig. 6, the current and voltage relations being shown in Fig. 7. Note that the secondary winding of each transformer is divided into two parts, each part wound for a voltage of $\frac{E}{3}$.

Transformer voltage = $\frac{2}{3}$ line voltage = $\frac{2}{3}$ E.

Transformer current = line current = 1.73 I.

Kilovolt-amperes in winding = $\frac{2}{3}$ E × 1.73 I = 1.155 EI.

Each secondary winding is therefore carrying 1.155 EI kilovolt-amperes in order to deliver EI kilovolt-amperes. Therefore, the single-phase rating of each single-phase transformer must be 1.155 times one-third the group rating if based on the current in the secondary winding or 1.0775 times one-third the group rating if based on the average of the currents in both the primary and secondary windings.

The foregoing can be summed up as follows:

1. *Three single-phase transformers, delta-connected*—The sum of the single-phase ratings equals the group rating.

2. *Three single-phase transformers, star-connected*—The sum of the single-phase ratings equals the group rating.

3. *Two single-phase transformers, V-connected*—The sum of the single-phase ratings is 15½ per cent. greater than the group rating.

4. *Two single-phase transformers, T-connected*—The sum of the single-phase ratings is 15½ per cent. greater than the group rating.

5. *Two single-phase transformers connected for three-phase to two-phase transformation*—The sum of the single-phase ratings is 15½ per cent. greater than the group rating if based on current in the windings connected three-phase, or 7¾ per cent. greater if based on the average of the currents in both the three-phase and two-phase windings. Two standard single-phase transformers for such service should have a total capacity 15½ per cent. in excess of the group rating.

6. *Three single-phase transformers with one winding connected in interconnected star*—The sum of the single-phase ratings is 15½ per cent. greater than the group rating if based on current in the windings connected in interconnected star or 7¾ per cent. greater if based on the average of the currents in both the interconnected star windings and the delta or star-connected windings. Three standard single-phase transformers for such service should have a total capacity 15½ per cent. in excess of the group rating.

Assuming that 300 kv.-a. are transformed in each of the cases given, the single-phase ratings of the individual transformers should be as follows:

1. 100 kv.-a. (3 required).

2. 173.5 kv.-a. (2 required).
3. 173.5 kv.-a. (2 required).
4. 100 kv.-a. (3 required).
5. 173.5 kv.-a. (2 required).
6. 115.5 kv.-a. (3 required).

Transformers are usually specially designed for the last two cases (5 and 6) and have increased capacity in the three-phase or interconnected star winding only. When operating such transformers single phase, the lighter winding determines the capacity of the transformer.

Ratings for Operation of Rotary Converters and Motors

In specifying transformers for the operation of rotary converters and motors, the apparent efficiency of the apparatus must be considered. In general, the apparent efficiency of a rotary converter may be assumed as 90 per cent. so that, for example, if a rotary converter has an output of 300 kw., a transformer capacity of 333 kv.-a. will be required for operating it. This relation of transformer capacity to rotary converter capacity must always be adhered to.

The apparent efficiencies of motors vary widely, depending upon the size, frequency and speed but will generally vary from about 75 per cent. to 90 per cent. It is customary to allow 1 kv.-a. transformer capacity for each horsepower of the motor, excepting for large motors where the transformer capacity should be calculated from the efficiency and power factor of the motor.

Voltage Classes

Transformers are divided, according to high-tension voltages, into the following twelve classes: 6600, 11,000, 16,500, 22,000, 33,000, 44,000, 55,000, 66,000, 77,000, 88,000, 100,000 and 110,000. While these voltages are considered standard, transformers may be wound for any reasonable high-tension voltage desired. In specifying a transformer with a high-tension voltage other than one of those given in the above classification, note particularly that each voltage class indicates the highest voltage for which any transformer belonging to that class can be designed. For example, if a transformer is to be designed for 46,000 volts, it will belong to the 55,000-volt class and not to the 44,000-volt class.

In specifying single-phase transformers to be connected in star on a three-phase circuit, the line voltage and not the transformer voltage determines the voltage class to which the transformer belongs. An exception may be made to this in case the neutral point is to be connected directly and permanently to ground. The transformer voltage (the voltage from the neutral point to the line) may then be considered as determining the voltage class.

Voltage Taps

To compensate for line drop, standard taps are provided on both high-tension and low-tension windings to give voltages approximately 3, 6 and 9 per cent. below normal. For example, if a transformer is wound for a normal ratio of 33,000 to 2200 volts, taps are provided to give additional high-tension voltages of 32,000, 31,000 and 30,000 and additional low-tension voltages of 2130, 2070 and 2000. With this arrangement, seven different ratios of transformation are obtainable, as shown in the following table:

16.5—1	16—1	15.5—1	15—1	14.5—1	14—1	13.5—1
36,300	33,000	33,000	33,000	32,000	31,000	30,000
2,200	2,070	2,130	2,200	2,200	2,200	2,200
	32,000	32,000	32,000	31,000	30,000	
	2,000	2,070	2,130	2,130	2,130	
		31,000	31,000	30,000		
		2,000	2,070	2,070		
			30,000			
			2,000			

From this it will be seen that if two transformers are connected to a transmission line, one as a step-up and the other as a step-down transformer, a total voltage drop of approximately 18 per cent. can be compensated for. Taking the case referred to above, with an impressed voltage of 2200 on the primary of the step-up transformer, and with the ratio of transformation of 1 to 16.5, the voltage delivered to the line will be 36,300, and by using the 13.5 to 1 ratio on the step-down transformer, a voltage of 30,000 impressed on the primary will deliver normal voltage of 2200 at the second terminals. That is, the two transformers together are able to compensate for a line drop of 6300 volts and still deliver normal voltage at the receiving end.

Parallel Operation

Transformers should never be connected together for parallel operation unless the individual transformers comprising the group are built under the same specification and are, therefore, alike. If it is ever necessary to operate unlike transformers in parallel, paralleling coils must be employed or impedance coils placed in the circuits of the transformer having the lower impedance. Paralleling coils are similar to balancing coils in construction and should be used in preference to impedance coils wherever possible, as they can be used in nearly all cases except when operating conditions are not permanent. For example, where one bank of transformers having certain characteristics is to be operated in parallel with two, three or more banks of transformers which are alike in themselves but differ in characteristics from the single bank, such a condition would require a paralleling coil with several taps, necessitating more or less expensive switching apparatus. In such cases, impedance coils are necessary.

(*To be concluded*)

Interconnected Large Power Systems—An indication of the line of thought which is being actively pursued in America can be gained from noting what is really happening in this country, in a territory where a network is being set up which is already within measurable distance of the ideal forecasted by Edison, Ferranti and Steinmetz—no small fry here, no duplication and multiplication of buildings, boilers, bunkers, boosters and busbars, but a few giant steam and hydraulic stations turning out their load according to a real schedule of conservation. . . . The economical operation of such systems as the Southern Power Co., a large network in itself, which is tied in with the Yadkin River Power Co., at Method, and is contemplating tying in with the Georgia Power Co., at Easley, and the latter with the Tennessee Power Co., at Rome, is a problem of considerable importance. The Southern Power Co. is operating approximately 350 miles of 100,000-volt lines as well as an extensive network of 44,000-volt and 11,000-volt lines, and the Yadkin River Power Co. has approximately 190 miles of 100,000-volt lines. The Georgia Power Co. will have approximately 200 miles of 100,000-volt lines, and the Tennessee Power Co. nearly 200 miles of 120,000-volt lines, as well as approximately 100 miles of 66,000-volt lines, making a total of nearly 800 miles of 100,000-volt lines in the four systems, and hundreds of miles of lower voltage lines, all tied in one vast system. The total distance by transmission line from Blewetts Falls to Nashville will be over 700 miles.—"Electrical Review."

GAS POWER DEPARTMENT

Gasoline-Engine Troubles and Their Symptoms

By A. L. Brennan, Jr.

Since nearly all gas- or gasoline-engine difficulties are indicated by some form of symptoms an operator should first of all become thoroughly familiar with his motor from this standpoint. Suppose an engine refuses to start; this may be caused by any one of the following:

1. Lack of fuel.
2. Flooded motor.
3. Water in cylinder or carburetor.
4. Dirty spark plugs or igniters.
5. Short- or open-circuit in the ignition system.
6. Poor compression.
7. Carburetor out of adjustment.

If the ignition and carburetion are known to be in good adjustment, yet the motor fails to start, this would indicate that some of the mechanical features are at fault. In this case the motor should be cranked over with all the relief valves closed in order to test the compression. If considerable resistance is encountered on the compression stroke without any binding, the compression is evidently all right. But if the piston moves up without meeting much resistance it is an indication that the compressed gas is escaping, which may be due to dry or scored cylinders, worn piston rings, faulty valves or valves out of time, partly open compression cocks or defective gaskets.

Ignition circuits are easily tested by removing the spark plugs, reconnecting the high-tension wires and placing the threaded portion of the plug on the cylinder; then place the switch in position and crank the motor over to the firing point of each cylinder and note if a bright spark appears between the points of the spark plugs. It should be remembered that a spark produced under ordinary conditions will have to be quite strong or it will prove inadequate when subjected to the high compression within the cylinder.

The only way to find out if a carburetor is properly adjusted is to make sure first that the controlling or mechanical features of the motor are in good order and that the ignition is all right; then attempt to start and note the symptoms.

Carburetors are an auxiliary feature of a gas engine; however, they do not generate the mixture independently of the motor as is commonly supposed, but are directly dependent upon the action of the piston. That is, on the intake stroke, the piston creates a partial vacuum within the cylinder which produces a suction through the carburetor, thus mixing the fuel and air together and drawing a charge into the cylinder. From this it is quite evident that any mechanical failures, such as leaky valves, etc., that interfere with good compression will also militate against good vacuum pulling; hence impaired carburetion.

When an attempt is made to start a motor and it refuses to go, it is generally advisable to resort to priming; that is, injecting a little fuel into the cylinders and again cranking the motor. If it then starts but stops, accompanied by back-firing, this is a positive sign that the mixture is too weak. On the other hand, if the motor continues in operation but runs in a sluggish manner it would indicate an over-rich mixture. When a motor operates in a regular manner, but does not develop high efficiency it may be due to:

1. Faulty carburetor regulation.
2. Fuel of low grade.
3. Inefficient or insufficient lubrication.
4. Poor compression.
5. Exhaust partly choked up.
6. Ignition out of time.
7. Coils (high tension) out of adjustment.

Sometimes an engine will not develop its full power, due to being "choked up," that is, the throttle has been open too wide when the load was applied and the engine has failed to clear itself. The remedy is to disengage the load and allow the motor to speed up several times.

Misfiring and other irregular operation may be caused by:

1. Faulty carburetion.
2. Water in the carburetor.
3. Dirty or cracked spark plugs.
4. Poorly adjusted coils.
5. Dirty contacts on the vibrator.
6. Weak batteries.
7. Loose wires.

When the misfiring is general in all cylinders, it may be due to water in the carburetor or a loose battery connection; but, on the other hand, if the skipping is confined to a single cylinder it is probably due to a dirty spark plug or a loose connection in the secondary circuit, for if it was caused by dirt in the carburetor, for instance, the other cylinders would miss as well.

When a motor slows down and stops, especially when accompanied by backfiring, it indicates a shortage of fuel; while sudden stopping more often results from a broken or disconnected wire, stuck timer points, trembler on coil stuck or in rare cases to sudden exhaustion of the fuel or to the needle valve becoming clogged.

Explosions in the exhaust are in some cases due to one cylinder misfiring and exhausting unburned gas into the exhaust pipe where it comes in contact with the flames from the other cylinders and ignites. It is also caused by faulty exhaust valves, or to a slow-burning mixture, inefficient spark or to an over-retarded spark.

Overheating is usually due to insufficient circulating water, retarded circulation, an over-rich mixture or to continued operation on a retarded spark.

Hissing sounds invariably indicate air or gas escaping under pressure. This may be due to loose or defective spark plugs or igniters, partly open compresion cocks or defective gaskets.

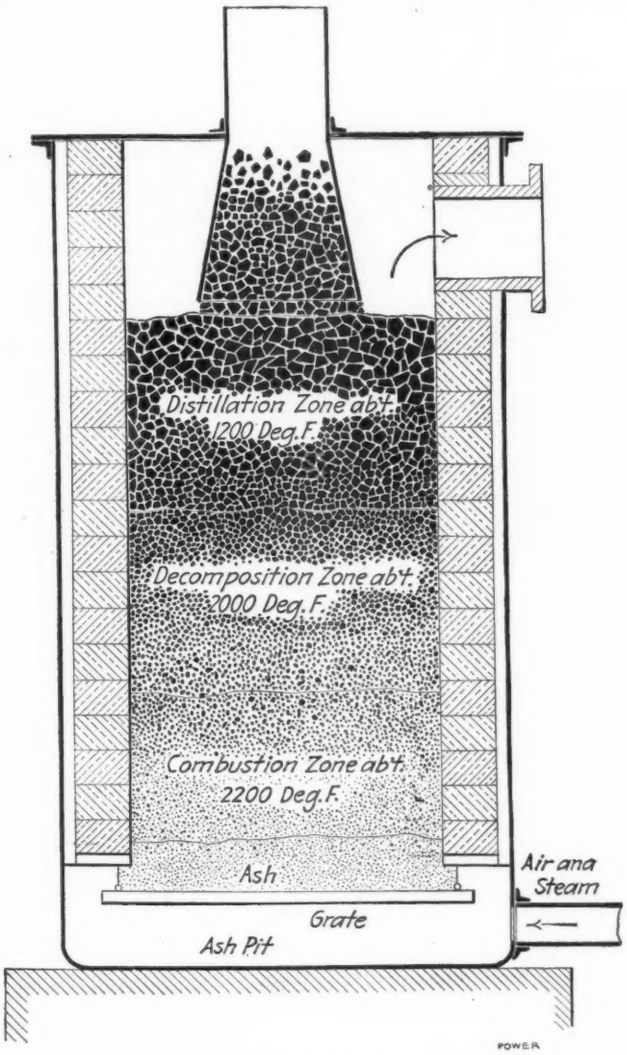
Knocking or pounding has two causes: mechanical and

functional. The first has reference to any lost motion that may exist in the bearings through wear or otherwise. The second has to do with the conditions that govern operation; that is, knocking is caused by ignition taking place too early or to an over-rich mixture. Carbon deposits sometimes become incandescent, causing preignition which is similar to knocking induced by an over-advanced spark.

Gas-Producer Chemistry

By G. W. Muench

For the successful operation of a gas producer it is desirable that the operator have a working knowledge of the chemistry of combustion. A thorough study of chemistry is not necessary as he needs only to get acquainted with four or five elements and a few of their compounds



Elementary Producer, Showing Various Zones

in order to get a good knowledge of the actions that take place inside a gas producer.

The elements with which to become familiar are carbon (C), hydrogen (H_2), oxygen (O_2), and nitrogen (N_2). The compounds that concern producer work are mainly water (H_2O), marsh gas (CH_4), carbon dioxide (CO_2), carbon monoxide (CO), and a number of heavy hydro-carbons, of which ethylene (C_2H_4) is the most important. These heavy hydrocarbons may be considered under the head of illuminants. Carbon compounds are so numerous that a study of these constitutes a separate branch of chemistry.

Hydrogen is not found free except in very small quantities. It is produced in gas making through the dissociation of water by high heat. Due to its extreme lightness the heat units in a cubic foot are not as high as many other gases, but the heat evolved in burning a pound of hydrogen is more than any other of the simple gases.

Oxygen forms 23 per cent. by weight and 20.7 per cent. by volume of the air. It is a supporter of combustion and when present in producer gas indicates that air has been drawn in from the outside through leaks. However, the absence of free oxygen does not necessarily prove that there are no leaks, as all the oxygen may have combined with carbon and hydrogen to form compounds.

Nitrogen forms about 77 per cent. by weight and 79.3 per cent. by volume of the air. It is inert and will not support combustion; that is, it enters the producer as nitrogen and leaves as nitrogen.

Carbon dioxide (CO_2) is a compound consisting of two parts of oxygen to one part of carbon by weight. It will not burn since it is itself the product of combustion. Therefore, it is undesirable in producer gas.

Carbon monoxide (CO) should always be a prominent constituent of producer gas, and should be kept as high as possible. It is formed by the incomplete combustion of carbon. This is the poisonous constituent of producer gas, and being odorless is hard to detect in the atmosphere when leakage occurs into the room.

Marsh gas, or methane (CH_4) comes from the distillation of oil or coal at fairly high temperatures, also from the slow decay of animal and vegetable matter under water. It is the principal constituent of natural gas, is easily handled in the gas engine, and has a high heat value. The higher the percentage of marsh gas in producer gas, the better.

Illuminants or heavy hydrocarbons are formed by the distillation of oils, coal, fats, etc. These hydrocarbons give the gas its lighting qualities. However, they are not present in producer gas in quantities worth considering.

Producer gas is a mixed gas consisting of carbon dioxide, oxygen, carbon monoxide, marsh gas, hydrogen and nitrogen, and occasionally traces of illuminants. It is formed by passing air and steam through a rather deep bed of fuel. The oxygen of the air unites with the carbon of the coal to form carbon dioxide. This takes place in the lower layers of the fuel bed. As this carbon dioxide leaves the combustion zone, as the lower zone is called, it passes up through the less highly heated zone above, called the decomposition zone. Here the CO_2 takes up more carbon to form CO. These reactions are expressed chemically as follows:

$$C + O_2 = CO_2 \ (\textit{Combustion zone})$$
$$CO_2 + C = 2\,CO \ (\textit{Decomposition zone})$$

The nitrogen remains unchanged and passes through the fire mixing with the CO.

The steam upon passing through the fire is broken up into its two constituents, oxygen and hydrogen, and the hydrogen, without any further change passes on with the mixture. The oxygen, however, unites with the carbon in

the decomposition zone to form CO. The changes taking place are expressed in this manner:

$$2\,H_2O + heat = 2\,H_2 + O_2$$
$$2\,C + O_2 = 2\,CO$$

The two may be combined and written

$$H_2O + C = CO + H_2$$

The upper layers of fuel are called the distillation zone. Here the gases in the fresh fuel are distilled out by the heat of the gases coming from the lower zones. In this zone the marsh gas (CH_4) is formed from the volatile matter of the coal. Some hydrogen may also be formed here. These different zones and their approximate temperatures are shown in the illustration.

Having followed the formation of the gas, the chemical actions that take place in the engine will now be considered. In the mixing chamber the producer gas is mixed with the proper amount of air; the mixture is then drawn into the combustion chamber and there exploded. During the burning of the gases at the beginning of the expansion stroke the following chemical changes take place:

$$2\,CO + O_2 = 2\,CO_2$$
$$2\,H_2 + O_2 = 2\,H_2O$$
$$CH_4 + 2\,O_2 = 2\,H_2O + CO_2$$

The nitrogen again undergoes no change. The principal constituents of the engine exhaust are, therefore, CO_2 and N_2 with water vapor, and usually a little free O_2.

The quantity of heat produced by the complete combustion of various gases has been found by experiment, and the heat value of a mixture may be found from the heat values of the component gases and the percentage of each. When speaking of the heat value of a cubic foot of gas it is necessary to specify the conditions of temperature and pressure as these determine the actual weight, and therefore the heat, contained in a cubic foot. This standard condition is a temperature of 62 deg. and an atmospheric pressure corresponding to 30 in. of mercury.

The most accurate method of determining the heat value of a gas is by means of a calorimeter. But as these instruments are expensive and require considerable laboratory experience, they are beyond the reach of the ordinary operator. If, however, the gas mixture has been analyzed and the heat values of its component gases are known, it is a very simple matter to figure its heat value and get results near enough for practical purposes.

As stated before, producer gas is a mixture of a number of other gases. Some of these gases burn, others do not. In producer gas CO, CH_4, and H_2 are combustible; CO_2, O_2 and N_2 are noncombustible. By experiment the heat values of these combustible gases have been ascertained. As analyses of gases are almost invariably given by volume, when figuring the heat value it is necessary to know the heat value per cubic foot of these various combustible gases. These values can be got from any reliable book on this subject. The following figures are representative for the heat values of the three gases, which make up the combustible part of producer gas. The heat value in British thermal units per cubic foot for hydrogen may be taken as 327, for carbon monoxide 324, and for marsh gas 1013. Having an analysis of the gas, and the foregoing heat values of the combustible constituents, the heat value of the mixture is figured by multiplying the heat value of each constituent by the fraction of a cubic foot of that constituent present in a cubic foot of the mixture and by then adding all the heat values.

For example, let it be desired to figure the heat value of the following producer gas, which is the analysis of a sample taken from a lignite suction producer: CO_2, 3.9 per cent; O_2, 1.3 per cent; CO, 16.8 per cent; CH_4, 8.5 per cent; H_2, 10.8 per cent; N_2, 58.1 per cent.

$$CO = 324 \times 0.168 = 54.5$$
$$CH_4 = 1013 \times 0.085 = 86.1$$
$$H_2 = 327 \times 0.108 = 35.3$$
$$Total = 175.9\ B.t.u.\ per\ cu.ft.$$

The calculation shows how simple it is to figure the heat value of a gas, knowing its analysis. With some study and practice during his leisure hours it is within the operator's reach to learn how to analyze the gas himself. It is a very simple process to at least find the amount of CO_2, O_2 and CO present, especially with an Orsat or similar apparatus. Although this analysis is not enough to figure the heat value it gives a clue to the condition of the fire in the producer.

The analysis should show only a very small amount of CO_2. An excess of CO_2 in producer gas is evidence of wasteful combustion inside the producer, of too much air, or of too little fire, or too hot a fire.

Free oxygen in producer gas is evidence of air mixed after the gas has passed through the fire. Free oxygen cannot come from the blast, as it would be converted into CO. Carbon monoxide should be the prominent constituent of producer gas, and the higher the percentage present the better. So small a percentage of CO would be due to the same troubles that cause excess of CO_2.

Too much hydrogen is likely to cause premature ignition in the engine, a trouble which may cause serious results; therefore, the percentage of hydrogen must be kept within limits. An excess is due to too much steam being drawn through the fire.

Marsh gas (CH_4) is a valuable constituent of producer gas, as it has a high heat value; the more present the better. But should an analysis of the exhaust gases show the presence of CH_4, there is a great deficiency of air, as the combustion is incomplete.

Sticking Fuel Valve

In the operation of an 80-hp. oil engine, using crude oil, considerable trouble was experienced during a cold snap by the fuel valve sticking. The trouble was thought to be due to the viscosity of the oil, and the following scheme for preheating proved both cheap and effective.

Four 16-cp. incandescent lamps were placed around the copper feed pipe in close proximity to the valves and thoroughly covered with asbestos. Practically all of the energy put into the lamps was thus converted into heat and transferred to the copper pipe, which in turn heated the oil as it passed through.

Bisbee, Ariz. G. S. WILLS.

A Speed of 3310 r.p.m. for an automobile motor was established by the automobile testing plant of the Worcester Polytechnic Institute, Mass. The conditions under which the tests were made approximated, as closely as possible, actual duty on the road. A dynamometer indicated that the 20-hp. rating of the motor, a Studebaker, was reached at 1120 r.p.m. At 1900 r.p.m., 30 hp. was developed, and at 3310, the maximum 36 hp., equivalent to a speed of 89 mi. per hr. At about 3300 r.p.m. the power curve began to drop.

HEATING AND VENTILATION

Heat Transmission with Pipe Coils and Cast-Iron Heaters under Fan Blast Conditions*

A series of 14 tests made at the Institute of Thermal Research of the American Radiator Co., on pipe coils of standard make, with the 1-in. pipe spaced on 2¾-in. centers, were discussed by the author and for comparison

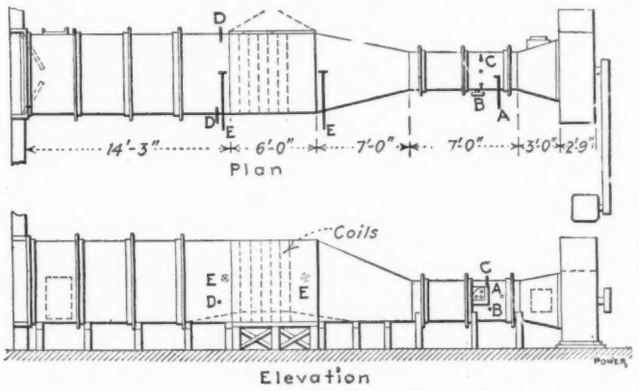


FIG. 1. PLAN AND ELEVATION OF TESTING APPARATUS

with the results equivalent values in Vento heaters were given. Tests were made on five sections, three sections and one section deep. Each section contained four staggered rows of pipe, making 193.1 sq.ft. of surface to each section, with a clear area for the passage of air of 17.06 sq.ft.

Steam was generated in cast-iron sectional boilers at pressures ranging from 9 to 18 lb. per sq. in. These pressures were reduced at the coils to a lower pressure ranging from 2 to 6 lb. per sq.in. The condensation was cooled and weighed in barrels. There was a steam separator in the high-pressure line and a similar one in the low-pressure line. Each coil was vented out of the return compartment on the same end of the coil where steam, return and bleeder connections were made. Each coil had individual valves on every connection. The bleeder header was looped into the return header. A steam-jacketed tank, with pressure equalized from the return header, served as a means of observing and regulating the flow of condensation out of the return pipe. The tests were started and stopped with the water level in the tank at a certain height, as shown by a string on the gage glass, so as to keep the bleeder loop sealed but leave the return header dry. The temperature of the steam was taken by a single calibrated thermometer and there was a similar thermometer on the return connection to each coil.

Before starting a test the coils were found to be hot all over. The variable-speed motor and dampers in the discharge piping provided means to obtain any desired velocity of air through the coils. Velocities were measured with a pitot tube and manometer, and these velocities agreed within an average of 6.65 per cent. of the velocities calculated from the amounts of condensation weighed. The velocities derived from the condensation values were used as test results since these results gave the total amount of heat taken out of the steam and therefore the maximum heating of the air. The velocities measured with the pitot tube were lower than those calculated out of doors from the face of the coils and a small amount of heat was radiated through the heater and the fan outlet,although all this sheet-steel work was covered 2 in. thick with hair felt. All the steam, return and bleeder headers and connections were also covered with hair felt.

With the pitot tube readings were taken at *A*, Fig. 1, on one horizontal, one vertical and two diagonal diameters, 10 readings on each diameter, thus making readings at 40 different points. These 40 readings were taken twice in an hour. The average velocity was calculated from these velocity-pressure readings and this velocity was reduced to its value for dry air at 70 deg. temperature and 29.92 in. barometer, which is used as a standard. At *D* is shown the location of the cold-air

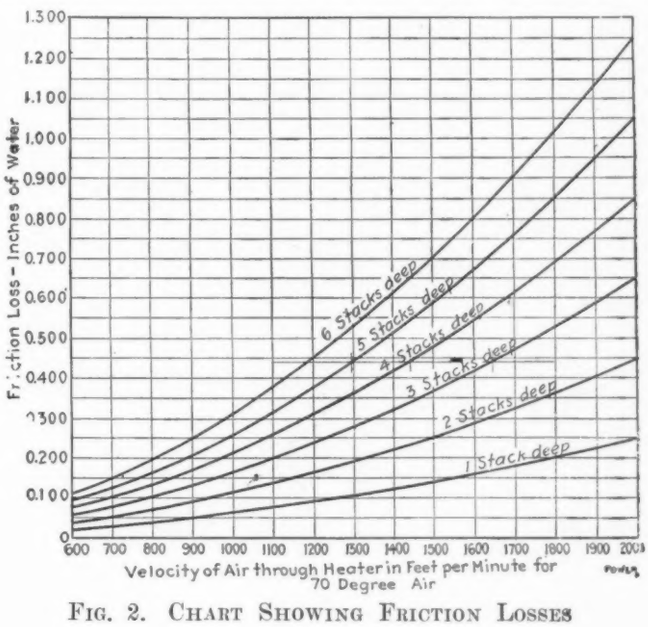


FIG. 2. CHART SHOWING FRICTION LOSSES

thermometers, at *E* the two static-pressure tubes, at *C* the five thermometers measuring the final temperature of the air, and at *B* the manometer registering the velocity pressure. The entire sheet-steel work from the cold-air inlet to the fan was made airtight, since the air was measured on the suction side of the fan.

Tables 1, 2 and 3 show the results of these pipe-coil tests and the equivalent values in Vento heaters. The Vento charts have been used to work out the universal performance of the pipe coils. To illustrate, take the results from Test No. 5 in Table 1 on five 4-row sections

*Abstract of paper read by L. C. Soule before the American Society of Heating and Ventilating Engineers, July 17-19, Buffalo, N. Y.

of pipe coils. Weather clear, barometer 29.95 in., temperature of steam in coils 223 deg.

Refer to the Vento friction chart, Fig. 2. Draw a horizontal line through a friction of 0.44 in. of water, which is the friction observed in test No. 5 for five sections deep of pipe coils at 1054 ft. per min. velocity. This line intersects three stacks deep of regular Vento sections on 5-in. centers at 1645-ft. velocity, four stacks deep at 1440-ft. velocity and five stacks deep at 1297-ft. velocity.

TABLE 1. COMPARISON OF 1-IN. PIPE COILS WITH 60-IN. REGULAR SECTION VENTO

PIPE COILS, 5 SECTIONS DEEP, 20 PIPES

Heater	Pipe coils	Vento	Pipe coils	Vento	Pipe coils	Vento	Pipe coils	Vento	Pipe coils	Vento	Pipe coils	Vento
Date of test, 1913	Jan. 8	.....	Jan. 8	.....	Jan. 9	.....	Jan. 9	.....	Jan. 9	.....	Jan. 9	.....
Number of test	3	.....	4	.....	5	.....	6	.....	7	.....	8	.....
Number of stacks or sections deep	5	4.038	5	4.121	5	4.055	5	4.022	5	4.038	5	4.036
Number of loops or pipes wide	25	14.37	25	14.25	25	14.65	25	14.88	25	14.15	25	14.24
Centers of loops or pipes, in	2.75	5	2.75	5	2.75	5	2.75	5	2.75	5	2.75	5
Friction, in. of water	0.095	0.095	0.31	0.31	0.44	0.44	0.384	0.384	0.25	0.25	0.186	0.186
Velocity of air, ft. per min. at 70 deg	484	670	860	1200	1054	1430	1000	1335	772	1084	662	925
Free area, sq.ft	17.06	12.32	17.06	12.22	17.06	12.57	17.06	12.78	17.06	12.15	17.06	12.21
Temperature entering air, deg. F	23.6	23.6	22.25	22.25	22.2	22.2	24.88	24.88	27.6	27.6	28.4	28.4
Temperature leaving air, deg F	128.2	128.2	110.4	110.4	104.4	104.4	108	108	117.3	117.3	121.4	121.4
Temperature rise, deg. F	104.6	104.6	88.15	88.15	82.2	82.2	83.12	83.12	89.7	89.7	93	93
Total condensation, lb	961.5	961.5	1431	1431	1639	1639	1576	1576	1312	13.12	1169	1169
Condensation per sq.ft. per hr., lb	0.996	1.035	1.482	1.526	1.697	1.725	1.632	1.647	1.359	1.436	1.21	1.27
Per cent. greater velocity with Vento	.....	38.5	.....	39.5	.....	35.7	.....	33.5	.....	40.5	.....	40
Total heating surface, sq.ft.	965.5	928	965.5	938	965.5	950	965.5	957	965.5	914	965.5	920
Per cent. surface required	100	96.1	100	97.1	100	98.4	100	99.1	100	94.7	100	95.3

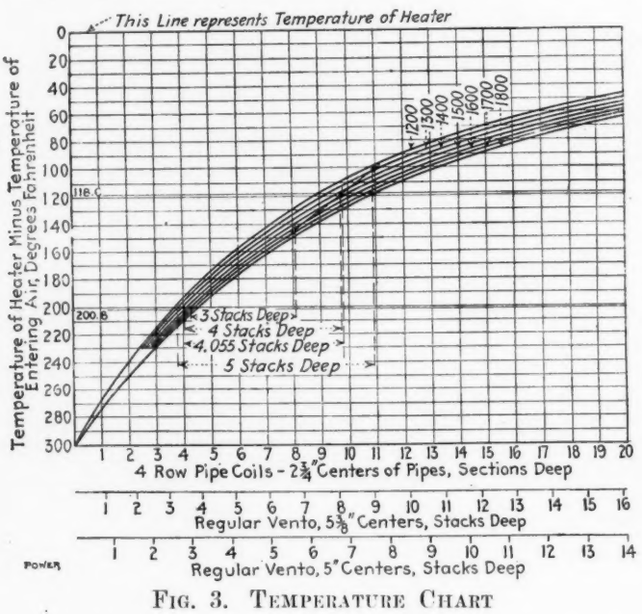


FIG. 3. TEMPERATURE CHART

Now refer to the Vento temperature chart, Fig. 3. The temperature of the steam in the heater for test No. 5 is 223 deg. Subtracting from this the 22.2 deg. temperature of entering air leaves 200.8 deg. Subtracting from 223 deg. the 104.4 deg. temperature of air leaving heater leaves 118.6 deg. Draw these horizontal lines of 200.8 deg. and 118.6 deg. temperature on the Vento temperature chart. At the points where the 200.8 deg. line intersects 1645, 1440 and 1297-ft. velocity, lay off from these points to the right, three deep, four deep and five deep, respectively, of Vento stacks of regular sections on 5-in. centers. From these new points extend vertical lines upward—each to its point of respective velocity. This gives three points which must be connected with a curve as shown. At the point where this curve intersects the 118.6-deg. line a velocity of 1430 ft. per min. is obtained. Extend a line downward from this point and another line downward from the point of intersection of 1430-ft. velocity with the 200.8-deg. temperature line and it will be found that the horizontal distance apart of these two vertical lines is equal to 4.055 stacks deep of Vento regular sections on 5-in. centers. This value for Vento is shown in Table 1 in the column just to the right of pipe-coil test No. 5 with which the Vento is compared.

Test No. 5 velocity through pipe coils of 1054 ft. per

TABLE 3. COMPARISON OF 1-IN. PIPE COILS WITH 60-IN. REGULAR SECTION VENTO

PIPE COILS, 1 SECTION DEEP, 4 PIPES

Heater	Pipe coils	Vento	Pipe coils	Vento	Pipe coils	Vento
Date of test, 1913	Jan. 14		Jan. 14		Jan. 14	
No. of test	9	.....	10	.....	11	.....
No. of stacks or sections deep	1	0.713	1	0.735	1	0.747
No. of loops or pipes wide	25	16.1	25	15.15	25	14.16
Centers of loops or pipes, in	2.75	5	2.75	5	2.75	5
Friction, in. of water	0.114	0.114	0.0325	0.0325	0.075	0.075
Velocity of air, ft. per min. at 70 deg.	1296	1600	637	835	922	1295
Free area, sq.ft	17.06	13.83	17.06	13.0	17.06	12.15
Temperature entering air, deg. F	30.4	30.4	33.6	33.6	36.75	36.75
Temperature leaving air, deg. F	48.3	48.3	56.7	56.7	56.56	56.56
Temperature rise, deg. F	17.9	17.9	23.1	23.1	19.81	19.81
Total condensation, lb	440.5	440.5	279.5	279.5	347	347
Cond. per sq.ft. per hr., lb	2.28	2.394	1.448	1.57	1.796	2.054
Per cent. greater velocity with Vento	....	23.5	....	31.0	....	40.5
Total heating surface, sq.ft	193.1	184	193.1	178	193.1	169
Per cent. surface required	100	95.2	100	92	100	87.5

TABLE 2. COMPARISON OF 1-IN. PIPE COILS WITH 60-IN. REGULAR SECTION VENTO

PIPE COILS, 3 SECTIONS DEEP, 12 PIPES

Heater	Pipe coils	Vento	Pipe coils	Vento	Pipe coils	Vento	Pipe coils	Vento	Pipe coils	Vento
Date of test, 1913	Jan. 16	.....	Jan. 16	.....	Jan. 16	.....	Jan. 16	.....	Jan. 17	.....
Number of test	12	.....	13	.....	14	.....	15	.....	16	.....
Number of stacks or sections deep	3	2.395	3	2.417	3	2.439	3	2.593	3	2.483
Number of loops or pipes wide	25	14.56	25	14.47	25	14.98	25	14.12	25	14.59
Centers of loops or pipes, in	2.75	5	2.75	5	2.75	5	2.75	5	27.5	5
Friction, in. of water	0.095	0.095	0.2075	0.2075	0.30	0.30	0.141	0.141	0.241	0.241
Velocity of air, ft. per min. at 70 deg	615	840	917	1260	1130	1500	703	990	987	1345
Free area, sq.ft	17.06	12.5	17.06	12.42	17.06	12.86	17.06	12.12	17.06	12.52
Temperature entering air, deg. F	43.5	43.5	43.3	43.3	45.8	45.8	46.9	46.9	51.8	51.8
Temperature leaving air, deg. F	104.3	104.3	96.3	96.3	94.4	94.4	106.6	106.6	100.8	100.8
Temperature rise, deg. F	60.8	60.8	53	53	48.6	48.6	59.7	59.7	49	49
Total condensation, lb	711	711	922	922	1042	1042	796	796	918	918
Condensation per sq.ft., per hr., lb	1.228	1.274	1.592	1.65	1.8	1.788	1.375	1.358	1.586	1.583
Per cent. greater velocity with Vento	.....	36.6	.....	37.5	.....	32.7	.....	40.9	.....	36.2
Total heating surface, sq.ft	579.3	558	579.3	559	579.3	583	579.3	586	579.3	580
Per cent. surface required	100	96.5	100	96.5	100	100.6	100	101.15	100	100.12

min., multiplied by 17.06 (square feet free area), gives air volume at 70 deg., and this volume divided by 1430 (feet per minute equivalent velocity through Vento) gives a free area of 12.57 sq.ft. This free area is obtained in 14.65 loops wide of 60-in. Vento on 5-in. centers of loops. From these data a total surface of Vento of 950 sq.ft., as compared with 965.5 sq.ft. of pipe coils to do the same work, is obtained.

Three of the tests indicated about 1 per cent. more Vento required than pipe coils, while in the other eleven tests the slight advantage was in favor of Vento.

By plotting all of the pipe-coil test results on the universal temperature chart, Fig. 3, it is found that they agree very closely. From this chart can be read off the performance of these pipe coils under any usual condition of steam pressure, entering air temperature, number of sections deep and velocity of air through the coils.

It was found from the tests, in comparing pipe coils (with pipes on 2¾-in. centers) with Vento heaters, that for the same friction there is 35 per cent. greater velocity through the Vento than through the pipe coils—when using one stack deep of regular Vento on 5⅜-in. centers of loops against each section deep of four-row pipe coils. Similarly, four stacks deep of regular Vento on 5-in. centers equal in friction effect five sections deep of four-row pipe coils when the velocity is 35 per cent. greater through the Vento. The foregoing comparisons in this paragraph apply to a condition of raising a given volume of air through the same range of temperature with the same frictional loss. Under these conditions of equality the amount of heating surface required will be the same with pipe coils as with Vento heaters.

Expense of Operating Heating and Ventilating Plants

At the Buffalo meeting of the American Society of Heating and Ventilating Engineers, H. M. Hart read a paper on the above subject, which was based on a report of a committee comprising H. M. Hart, chairman, N. L. Patterson and G. L. Hubbard, to the Illinois chapter. Residence heating, apartment buildings and school buildings were given brief mention, but due to various reasons, there was a wide variation in the figures, so that these data will be omitted.

For office buildings Mr. Hubbard gave the following brief outline of his general practice in estimating the cost of operation of the heating apparatus.

The smaller the plant the greater the cost of heating will be per square foot of radiation, largely on account of the cost of labor. Assuming that the radiation shown on the plans is the correct amount for properly heating the building between the limits of outside and inside temperatures, and that the ventilating equipment is also correctly proportioned, the first thing to determine is the amount of steam that will be used in producing the heat necessary to warm the building. For this it is necessary to estimate the percentage of the total radiation that will be in use continuously and the length of the heating season. For example, in the usual office building in Chicago, with direct radiation in all stories above grade and with ventilating equipment for all stories below grade, he assumed that 65 per cent. of the direct radiation would be in use for the full 24 hours per day for a period of 7 months and 50 per cent. of the fan-coil surface for 10 hours per day for 6 months. With a condensation per square foot of direct radiation of ¼ lb. per hr. and per square foot of fan-coil surface of 1½ lb., the total amount of steam required per year can readily be determined.

Assuming that with the ordinary coal used an evaporation of 6 lb. of water per pound of coal can be obtained under all conditions of load and firing, day in and day out, the amount of coal required during the heating season can be obtained by dividing the steam used by 6. This reduced to tons and multiplied by the cost per ton (say about $2.60 under the usual conditions in Chicago), will give the cost of coal per year. Ash removals should be about 10c. per ton of coal, and the makeup water required should be about 5 per cent. of the total steam per year. The only other item to be considered is the labor, and this must be determined in each case on its own merits. There must, of course, be a man in charge, and as he will have to devote a large portion of his time to work in parts of the building outside of the boiler room, it is probably correct to charge one-third of his salary to the heating plant. In addition, there must be firemen and coal passers enough to handle the fires properly, remove ashes, and see to the normal operation of the boilers, and one or two handy men to do the odd jobs that are always requiring attention.

The foregoing make up all the actual outlay required to produce the steam for heating, but it is customary also to consider depreciation, repairs, etc., which amount to 12 per cent. of the original cost of the plant.

If the pumps required by the plant and the fans are steam-driven and the steam is turned into the heating system no special account need be taken of the amount used, but if they are motor-driven a further sum must be added to the foregoing cost: that for the electric current consumed, whether this is obtained from a plant in the building or from a central-power plant.

In reference to actual practice two different buildings of about the same type and size were taken, one having its own power plant, and the other operating a heating plant only, and the following data given.

In a modern office building equipped with a vacuum system, temperature control, low-pressure boilers and smokeless furnaces, the theoretical fuel consumption is as follows:

The direct radiation is 60,850 sq. ft. The average outside temperature for the seven heating months of 1911 and 1912 was 33.6 deg.; therefore, the theoretical number of hours that radiators would be turned on would be

$$\frac{70 - 33.6}{80} = 0.455$$

and 45½ per cent. of 210 days × 24 hr. would be 2293 hr. Therefore the steam required for heating would be

$$\frac{60{,}850 \times 225 \times 2293}{961} = 32{,}668{,}091 \text{ lb.}$$

To this should be added the loss through piping which is covered with 85 per cent. magnesia. This has been estimated to be 3 per cent. of the total, which would make the total loss by radiation 33,648,133 lb.

For ventilation there are the following units:

One unit delivering 32,420 cu.ft. per min. at an average rise of 26.4 deg. for 20 hr. per day.

One unit delivering 18,400 cu.ft. per min. at an average rise of 28.4 deg. for 20 hr. per day.

One unit delivering 33,860 cu. ft. per min. at an average rise of 51.4 deg. for 10 hr. per day.

One unit delivering 19,500 cu.ft. per min. at an average rise of 76.4 deg. for 10 hr. per day.

The steam required for the above service would be as follows:

$$\frac{\text{Rise}\quad \text{Min.}\quad \text{Hr.}\quad \text{Days}}{32{,}420 \times 26.4 \times 60 \times 20 \times 180} = 3{,}497{,}717 \text{ Lb. of Steam}$$

$$\frac{18{,}400 \times 28.4 \times 60 \times 20 \times 180}{55 \times 961} = 2{,}135{,}521$$

$$\frac{33{,}860 \times 51.4 \times 60 \times 10 \times 180}{55 \times 961} = 3{,}556{,}212$$

$$\frac{19{,}500 \times 76.4 \times 60 \times 10 \times 180}{55 \times 961} = 3{,}044{,}147$$

This makes a total of 12,233,597 lb. of steam for ventilation, which, added to that required for heating, makes a total of 45,881,730 lb. of steam. When burning screenings and evaporating 6 lb. of water per pound of coal this amount of steam would require

$$45{,}881{,}730 \div (6 \times 2000) = 3823 \textit{ tons.}$$

The actual fuel consumption per month was as shown in Table 1.

TABLE I. ACTUAL FUEL CONSUMPTION IN OFFICE BUILDING.

	Theoretical Tons	Actual Tons	Outside Temperature, Degrees	Average Wind, Miles
October	248	301	53.3	12.8
November	542	573	35.4	16.9
December	547	468	35.0	14.4
January	783	913	11.9	14.2
February	759	656	21.8	14.4
March	640	661	28.8	13.5
April	338	286	48.8	16.5

It will be noticed from this table that, during the months of November and December, the temperature was about the same, but the wind velocity decreased about 15 per cent. and the fuel consumption about 18 per cent. The difference between the months of April and October, of course, is not consistent, but as the engineer had no means of weighing the coal as it was put into the boilers the figures given per month might not be absolutely correct. The cost of operation of this heating plant is given in Table 2.

TABLE 2. COST OF OPERATION OF HEATING PLANT

Coal, 3858 tons at $2.37	$9,143.46
Removing ashes	554.00
Oil, waste and packing	160.00
Repairs	100.00
Labor	4,500.00
Electric current for vacuum and boiler-feed pumps	429.00
Water, approximately	200.00
Interest and depreciation, 10 per cent	2,892.00
Total	$17,978.46

The actual cost of producing steam is

$$\frac{1000 \times \$17{,}978.46}{3858 \times 6 \times 2000} = 38.8c. \textit{ per } 1000 \textit{ lb.}$$

and if the fuel for water heating were added in there would be an additional expense of $2883 for coal and $174 for removing ashes, making the total expense per year $21,035.46. This would bring the cost of steam per 1000 lb. down to

$$\frac{1000 \times \$21{,}035.46}{5074 \times 6 \times 2000} = 34.6c.$$

In another building, almost a duplicate, having its own electric generating plant and hydraulic elevators, the heating load would be about as follows: 68,000 sq. ft. direct radiation, requiring

$$\frac{68{,}000 \times 225 \times 2293}{961} = 36{,}506{,}660 \textit{ lb. of steam}$$

The pipes were covered with molded asbestos and the loss through them may be estimated at 4 per cent., which would bring the load up to 37,966,926 lb.

For heating water the load may be estimated to be about the same as in the previous building, which required 14,600,000 lb. of steam, making a total of 52,566,926 lb. The cost of operation is given in Table 3.

TABLE 3. COST OF OPERATION OF POWER AND HEATING PLANT

6275 tons No. 4 washed nut at $3	$18,825.00
Removing ashes	890.00
Oil, waste and packing	470.00
Water	2,407.00
Lamp renewals	486.00
Labor	9,320.00
Interest and depreciation, 10 per cent	7,000.00
Total	$39,398.00

To obtain the cost of steam for heating, the deductions given in Table 4 must be made.

TABLE 4. DEDUCTIONS TO BE MADE TO OBTAIN COST OF HEATING

For 644,742 kw.-hr. generated:		
Fuel at 49 lb. steam per kw.-hr	$6,551.00	
Water	249.00	
Lamps	486.00	
Ashes	310.00	
Oil, waste and packing	100.00	
Labor	1,884.00	
Interest and depreciation	3,000.00	
Total	$12,580.00	$12,580.00
For Elevators:		
Coal	$7,712.00	
Water	257.00	
Ashes	364.00	
Oil, waste, etc	300.00	
Labor	2,700.00	
Interest, etc	1,000.00	$12,333.00
Total		$24,913.00

Then $39,398 — $24,913 = $14,485 would be the additional cost for heating. If this were taken at the same cost rate as the previous building, the cost of heating would be 52,566,926 lb. of steam at 34.4c. per 1000 lb., or $18,083. Therefore, the saving on the cost for heating is

$$\$18{,}083 - \$14{,}485 = \$3598$$

However, this does not represent the actual saving shown by the operation of this plant. The saving would be as follows:

Cost of heating without plant	$18,083.00
Revenue for kw. sold	25,855.00
Revenue for kw. for public lighting	9,928.00
Cost of elevator service	12,333.00
Total	$66,199.00
Less cost of operation	39,398.00
Saving	$26,801.00

❖

New Electric Interurban Railway—Service on the Oakland, Antioch and Eastern Railway was initiated September 3 when a schedule of eight trains daily between Sacramento and San Francisco became effective. On August 30 a two-car train made the run of 91 miles in 3 hr. and 10 min. The J. G. White Companies were represented on this trip by members of the San Francisco office.

The Shepard Pass tunnel, built for this road, and located about 5 miles from Oakland, is nearly two-thirds of a mile long, the longest thus far constructed in this country for an interurban electric railway. The new road is equipped with electric locomotives and steel cars including Pullman and observation cars.

The distribution system from the railway substations is 1200-volt direct current with overhead catenary construction.

The power is supplied by one of the large hydro-electric companies whose lines intersect the railway.

EDITORIALS

Cheaper Power—Reduced Manufacturing Costs

The revised tariff rates shortly to become effective are bound to have considerable effect upon the manufacturing industries of the country. Whether the results are to be beneficial or otherwise depends largely upon the foresight and enterprise of manufacturers in meeting increased competition, etc. In other words, the manufacturer who is to feel the increased competition, more than ever before, must decrease his manufacturing costs, among which one of the main expenses is that for power.

The small manufacturer will undoubtedly be the greatest sufferer and the logical solution, in any case, is increased production to compensate for decreased rate of profit. This is made possible by consequent increased markets. The manufacturer who is brought into competition with foreign products can only successfully meet that competition by reductions in the market price of his own output, which means that he must not only sell cheaper than formerly but he must and will sell more, decreased prices invariably creating increased demands. The manufacturer who is not affected by the introduction of foreign products will find an increased market for his goods, for statistics show that increased importation invariably creates an equal or greater export trade. The manufacturer who in the past has successfully entered the foreign markets will then find his export business rapidly increasing. Increased manufacture must therefore result from reduction in the tariff rates if the American manufacturer is to hold his own.

Increased manufacture will necessitate more factories, increasing the capacity of existing factories or the taking of other means of increasing output as well as reducing manufacturing costs. Building new factories and enlarging old ones will involve the investment of considerable capital, capital for which undertakings will be difficult to obtain until industrial and manufacturing operations recover from the readjustment of conditions that must be expected; i.e., until the industrial and manufacturing situation is once more stable.

Factory output could be increased without additional investment for power if continuous operation were possible. The average factory power house is now in productive operation only about ten hours a day. This is true of the coal-burning plant. Such a plant can be far more economically operated continuously than with the present long and regular periods of productive inactivity.

A power house that is productively operated twenty-four hours each day consumes more coal than one that is inoperative more than half the time, the cost of attendance is greater as is also the expense for oil, waste, supplies, etc., but the plant burden—depreciation, repairs, interest on the investment for land, building and equipment, insurance and taxes—is no greater, or very little greater than when the hours of operation are but ten a day; the result being that the yearly total cost of power in a plant operating twenty-four hours per day and one operating ten hours per day is seventy per cent. (in a plant of one hundred horsepower or so) to fifty per cent. (in a plant of several thousand horsepower capacity) more in the former than in the latter, although nearly two and one-half times as much power is available.

With coal at the average price, that means that the total cost of power per horsepower per hour (brake horsepower at the engine) in the small plant would be reduced from about two and five-eighths cents to about one and seven-eighths cents and in the large plant from about somewhat over one-half cent to not much more than one-third cent. Such economical power production would be even greater than that obtainable by doubling the size of the power house—doubling its ten-hour per day capacity by duplicating the power house. A saving of from one-sixth to three-fourths cent per horsepower, depending upon the size of the plant, would represent a considerable in the cost of manufacturing.

The manufacturer's bill for lighting would be considerably increased by adopting the twenty-four hour day, it is true, but the labor charge need not be appreciably increased by operating three eight-hour shifts (not overtime work) neither would the cost of raw materials be augmented. The latter, in fact, being reduced owing to the better prices obtainable in buying in so much larger quantities than formerly.

⁂

Boiler Efficiency

An inexplicable confusion seems to exist as to the basis upon which to calculate boiler efficiency. The 1899 report of the American Society of Mechanical Engineers committee on the "Revision of the Society Code of 1885 Relative to a Standard Method of Conducting Steam Boiler Trials" contains an ambiguity which, if its rare mention is any criterion, has gone undetected for more than ten years by a large majority of the engineering profession. POWER has made editorial mention of it, but, so far as we know, the society has never officially made a correction.

This error appears in rule XXI which states: "Two methods of defining and calculating the efficiency of a boiler are recommended. They are:

"1. *Efficiency of the boiler* =

$$\frac{\textit{Heat absorbed per lb. of combustible}}{\textit{Calorific value of 1 lb. of combustible}}$$

"2. *Efficiency of the boiler and grate* =

$$\frac{\textit{Heat absorbed per lb. of coal}}{\textit{Calorific value of 1 lb. of coal}}$$

All of which is just about equivalent to stating that two equals four, for the right side of equation 1 is exactly equal to the right side of equation 2, while the left sides obviously are not equal.

If 80 per cent. of the heat value of the combustible were absorbed the efficiency, according to equation 1, would be

$$\frac{80}{100} \times 100 = 80 \textit{ per cent.}$$

And if the combustible constituted 90 per cent. of the coal, the efficiency according to equation 2 would be

$$\frac{80 \times 90}{100 \times 90} \times 100 = 80 \textit{ per cent.}$$

the same as before.

From a subsequent paragraph of the committee's report is apparent that what was really meant was

1. *Efficiency of the boiler* =

$$\frac{\textit{Heat absorbed per lb. of combustible burned}}{\textit{Calorific value of 1 lb. of combustible}}$$

and

2. *Efficiency of the boiler and grate* =

$$\frac{\textit{Heat absorbed per lb. of coal fired}}{\textit{Calorific value of 1 lb. of coal}}$$

The Power Tests Committee's report presented at the 1912 annual meeting contains an improved method of calculating efficiency.

This report states:

"The efficiency of a boiler is the ratio of the amount of heat which it absorbs and uses in making steam to the amount of heat furnished to it.

"The heat which is furnished to the grate is all the heat contained in all the coal as fired.

"The heat furnished to the furnace and boiler is that in the coal which is *burned*. If a part of the good coal drops through the grate the boiler and furnace should not be charged with its heat.

"The Committee on Tests of the American Society of Mechanical Engineers, therefore, recommends the consideration of two efficiencies.

"The efficiency of the grate would equal the first of these formulas divided by the second or

$$\frac{\textit{Lb. of combustible burned}}{\textit{Lb. of combustible fired}}$$

"1. *Efficiency of boiler, furnace and grate* =

$$\frac{\textit{Heat absorbed per lb. of coal fired}}{\textit{Calorific value of 1 lb. of coal}} =$$

$$\frac{\textit{heat absorbed per lb. of combustible fired}}{\textit{calorific value of 1 lb. of combustible}}$$

"2. *Efficiency of boiler and furnace* =

$$\frac{\textit{Heat absorbed per lb. of combustible burned}}{\textit{Calorific value of 1 lb. of combustible}}$$

This is a great improvement on the older definitions of and formulas for efficiency and is, perhaps, as comprehensive as possible, consistent with simplicity.

One objection that might be raised is that no allowance is made for the moisture content of the coal as fired which, with some grades, runs rather high. It is not just to charge against the furnace and boiler the loss caused by the heat used to evaporate and superheat the moisture contained in the coal. This objection may generally be more technical than practical but it is legitimate, nevertheless.

Another weakness of existing or proposed standards of boiler efficiency is that no consideration is taken of the human factor, which has a very strong influence indeed upon results obtained, especially with hand-fired equipment. It does not seem feasible, however, to devise a formula to take this human element into exact account. However, in comparing published figures on efficiency it is well to bear the possibilities of the human factor carefully in mind and refuse to concede that one construction or type of apparatus is superior to another simply because a better showing is indicated by the conventional methods of estimation. An endeavor should first be made to analyze the report for the purpose of determining just how great the human element has influenced the results secured.

Mr. Williams' Exceptions

In a letter appearing on page 444, Arthur Williams takes exception to the opinions expressed in our editorial on the Hall of Records Test. If we were mistaken as to the purpose of the test we are glad to be set right, but this all the more strengthens our contentions, for if the real purpose, as Mr. Williams says, is to determine whether it is better for the city to continue operating that particular plant or shut it down and purchase light, heat and power, then in the latter event the fixed charges will go on just the same and should be added to the cost of the purchased product.

Without wishing to dispute Mr. Williams, however, we are still of the opinion that the outcome of the present test will have an indirect, if not a direct, bearing upon the city's policy as regards municipal plants in general.

The main contention of our editorial was that the city should have conducted its own test and not jointly with the New York Edison Co., thus not only taking the latter into its confidence and giving it a voice in determining the fixed charges, but also giving it daily logs of the test.

We expressed our explicit faith in those actually conducting the test and did not question the scientific standing of the engineers on the advisory board; nevertheless, we should expect the New York Edison Co.'s representatives to push their client's interests first, last and at all times as faithfully as in the past.

As to Mr. Williams' views on fixed charges we have only to remind him of his remarks at the joint meeting of the American Society of Mechanical Engineers and the American Institute of Electrical Engineers on Mar. 10, 1911. We have not seen evidence of any change in his views since then.

The Commission of Labor of New Jersey informs us that as there was no appropriation made with which to organize the Steam Engineers and Boiler Operators' License Bureau in the time specified by law, no definite time limit can be set within which application must be made to obtain an engineer's or fireman's license without examination. Nothing can be done by the New Jersey Department of Labor relative to licensing engineers and firemen until funds are appropriated and the Bureau organized. When these have been accomplished mention will be made in these columns.

We tried to make it plain in our Aug. 19 issue that the Department of Labor does not give out application blanks because it has none to give. Nevertheless several have written to that Department requesting blanks and mentioning our article.

READERS WITH SOMETHING TO SAY

Safeguarding Sumps in Power Stations

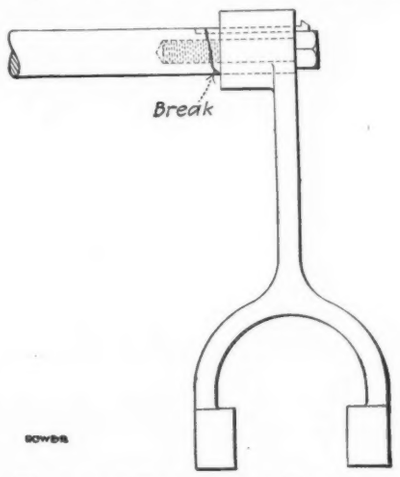
The importance of safeguarding sumps in power stations is shown by an accident that occurred at a lighting plant in a Western city. The pump room, constructed below the level of the ground and lighted by windows only, was drained by an uncovered sump, located at one of the corners of the room, which did not receive much light. The equipment of the pump room included three feed-water heaters, which had been shut down while a thermometer was being installed in the suction of the feed pump. During the progress of the work large quantities of hot water escaped and flooded the floor of the room, as the sump pump was necessarily stopped. The men walked upon bricks which had been placed on the floor. It was necessary to pass near the sump to reach one of the heaters, and the engineer, who had just put the heater in service again, slipped, in some unaccountable way, from the loose bricks and fell into the sump filled with hot water. The accident would not have occurred if the sump had been covered, and it would probably not have happened if the pump room had been lighted by electricity.

ERNEST F. LEARNED.

Watertown, Mass.

Repairs to Broken Rockshaft of Pump

The sketch shows where the rock shaft of a duplex pump was broken. Having to make a quick repair I proceeded as follows: Taking the rock shaft out I tapped a ¾-in. bolt to pass easily through for part of its length as shown. After putting the shaft back in place I tightened up on the bolt drawing the broken parts together, and then drove in the long key, which extends through the rocker-arm and into the rockshaft. The pump has been running this way for more than a year and gives satisfaction.



HOW THE BREAK WAS REPAIRED

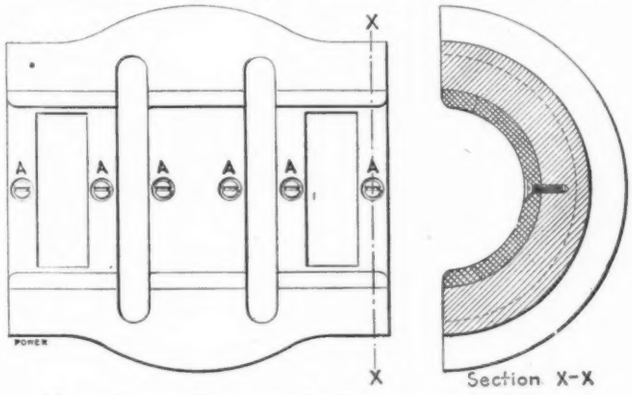
O. R. MASON.

Chicago, Ill.

Fastening Loose Babbitt

The following is a good way to fasten babbitt that has become loose in the bearing or cap:

Drill and countersink a hole through the babbitt and into the metal of the cap or bearing. Make the counterbore large enough so that the head of the screw will be below the babbitt, as shown.



HOW LOOSE BABBITT IS FASTENED TO BEARING

In repairing a large cap, such as shown, put in the screws indicated by *A*, then when these have been set up hard, drill more holes and distribute the screws according to the area of the babbitt to be held.

J. G. KOPPEL.

Montreal, Can.

Repairing a Pump Piston

Recently I was asked by the foreman machinist of our plant to help repair an old pump for tank service, the pump in question had been discarded by a former engineer as being unfit for use, and was stored away in a damp basement for years. It is a single low-pressure pump, and we found on examining it that outside of rust and dirt, the damage was in the form of a hole worn completely through the follower, evidently caused by allowing the pump to run with the studs and springs loose in the bottom of the piston.

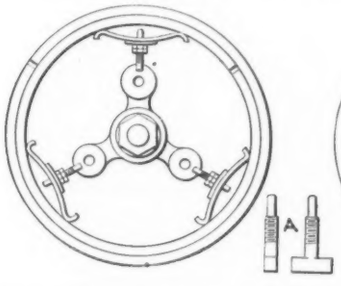


FIG. 1. FOLLOWER PLATE

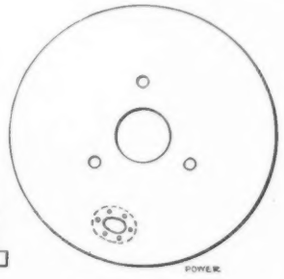


FIG. 2. PUMP PISTON

Fig. 1 shows the type of piston and Fig. 2 shows the follower and the location of the hole at the bottom.

We cut a piece of $\frac{3}{16}$-in. flat iron to the shape shown

by the dotted lines, Fig. 2, and riveted it to the outside of the follower. It was made tight by calking. To prevent the studs and springs from falling down, we made new studs like A, Fig. 1, with tee instead of square heads; the width of these heads was made a close fit in the slots in the spider, and the heads were made as long as possible without interfering with the follower bearing against the spider. This pump now gives satisfaction and the foreman and I are thinking that we have saved our company the price of a new pump.

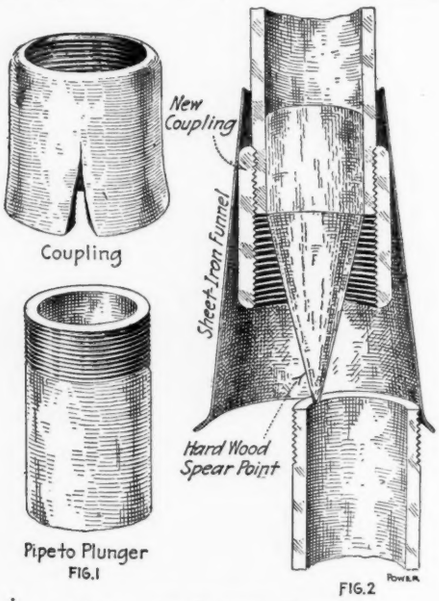
L. Johnson.

Exeter, N. H.

Removing Plunger from Deep Well

Readers may be interested in a method made use of in recovering the plunger and foot valve from a deep nonflowing artesian well.

During the process of overhauling and making general repairs to a deep-well pump of the vertical type used in this nonflowing artesian well, it became necessary to remove the rod, plunger and foot valve. After removing the large bolts at the base of the pump and swinging it from its position over the well, the pump rod, which was of extra heavy iron pipe, the lengths of which were connected by iron-pipe couplings, was drawn out through the 4-in. casing of the well.



Parts of Casing Finder

Upon removing the rod it was found that it had parted at a coupling near the plunger and that the plunger was still in the working barrel at the bottom of the well. Examination showed the coupling at the end of the rod to be much worn on one part from rubbing on the well casing. The extra strain on it, caused by the attempt to draw the plunger through the unused part of the working barrel had probably caused the coupling to split open at one end, see Fig. 1, and come up with the rod, leaving the plunger behind.

Instead of fishing for the remaining part of the rod with grappling hooks, as is commonly done in cases of this kind, the pump rod was again lowered into the well after being fitted with a new coupling, the threads of which had been made so as to insure an easy fit to the threads of the old pipe. A sheet-iron funnel and a hard-wood spear point were used to help guide the new coupling to its position for making up with the old pipe, as shown in Fig. 2.

The funnel at its open end was made only slightly less in diameter than the casing of the well; so that if the lower end of the pump rod were against the casing of the well at any position, the funnel when passing over it would cause it to be thrown toward the center, where the hard-wood spear point, being somewhat shorter than the funnel, would engage in the pipe and because of its tapering shape would throw the rod into the right position for making up with the coupling of the upper portion of the rod.

The device proved very successful and at the first attempt the threads of the coupling engaged those of the lower part of the rod and after a few turns with a wrench, the rod, with the foot valve and plunger, was drawn from the well without further trouble.

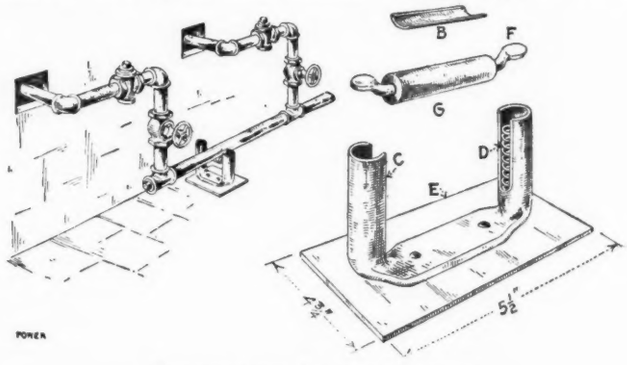
H. E. Brookman.

Trenton, N. J.

Expansion Support for Blowoff Line

In the plant where I am employed the boiler settings were all rebuilt. The boilers, six in number, are 150-hp. horizontal return tubular, each provided with a 3-in. blowoff pipe. The blowoff line is exposed, running along the floor and raised about 3½ in., see Fig. 1, to permit the boilers to settle, while drying out. This line could not be braced solid on account of the boilers settling and breaking the connection, so the following expansion brace was made.

A piece of ¾-in. pipe, 15 in. long, was punch-marked and ⅜-in. holes drilled through 3½ in. from each end, so that a piece shaped as *B* could be taken out of each end of the pipe as at *C* and *D*.



Support for Blowoff Pipes

The pipe was then heated to the proper temperature and flattened out at *E* in the center and the ends bent at right angles as shown. The pipe was next mounted on a base of iron ⅛x4½x5½ in. and two springs of the proper tension 4 in. long and $\frac{11}{16}$ in. in diameter inserted in the ends of the upright pieces *C* and *D*. The crossbar *F* is made from ½-in. iron with the ends flattened and rounded to $\frac{11}{16}$ in.; it is provided with a roller *G*, ⅝x2½ in. long. This bar rests on a spring and supports the main blowoff line.

C. R. Van Keuren.

Famosa, Calif.

QUESTIONS BEFORE THE HOUSE

Connecting Waterwheel and Engine

Under "Inquiries of General Interest" in Aug. 19 issue, I noted one in regard to connecting waterwheels to engines. From the experience of our company with the same sort of equipment, I would suggest that it is very inadvisable to connect waterwheels and steam engines together without a governor on the waterwheels as well as on the engine to prevent racing should the load become less than the capacity of the waterwheel. If the two prime movers are necessary at all times, the governors on the waterwheels need be only of such type as to prevent them reaching a dangerous speed. It seems preferable, if the two prime movers are driving electric generators, to connect them electrically if generating alternating current, rather than by a clutch of any form. Clutches are often unavoidable, where the combined power is to be delivered to a single shaft, but they are often troublesome to such an extent as to drive one to despair.

J. E. HARLOW.

Dixon, Ill.

The Failure of a Hot-Water Tank

From the description of the defects related by R. Cederblom, page 139, of the July 22 issue, as occurring on the head of a hot-water tank that came under his observation, I should say that he may be practically certain that the initial cause of the trouble was a breathing action, similar in character to that which produces the so called lap crack in boilers.

Any tendency toward pitting or corrosion along definite lines is almost invariably produced by some disturbance of the fibres of the metal along such lines, the disturbance preventing the formation of scale which would protect the metal from the corrosive action of the contents. When disturbance occurs, and fresh unprotected surfaces are constantly presented for corrosive action, the rate of action is unbelievably fast. Water that under other circumstances has no corrosive effect, will under such conditions cut a groove in a remarkably short time.

In boiler practice, grooving near the turn of the flange on flat heads of domes, if not braced, is very common. When the two-flue boiler was in general use, grooving at the girth seams was a common occurrence. These boilers were long and of small diameter, and their weight was usually carried at the front and rear ends. With a boiler of 42-in. diameter and 36 ft. long (which was an average size) there was considerable movement endwise, due to expansion, and the sag produced by the weight of boiler and contents, and the effect of this movement was concentrated at the girth seams, and produced girthwise grooves around the bottom half of the boilers on the inner surfaces of the sheets, near the edge of the girth-seam lap. The bumped heads of domes and drums are not immune to such action, because the standard bump does not produce a head that tends to retain its shape with the application of internal pressure, and in many cases the movement is sufficient to produce rapid growing.

Bracing a flat head by means of angle iron, as shown by Mr. Cederblom, is not satisfactory, unless the angles are carried right up to the turn of the flange, or better still, have the ends of the angle iron fitted to the sweep of the flange, but even under such circumstances this style of bracing is not suitable for high-pressure work.

J. E. TERMAN.

Hartford, Conn.

Factory Fire Chiefship

Referring to the editorial on the above subject on page 238 of the Aug. 12 issue, you undoubtedly know, if these factories would put in automatic sprinklers, the fire chief would be wholly supernumerary. Incidentally, there would be no loss of life, no panic, no fire loss to speak of, under any circumstances; and the reduction of insurance premiums would pay for the sprinklers within four or five years.

When commenting on such a disaster as that at Binghamton, do you not think it worth while to consider a few of the points above mentioned?

SIDNEY S. KOON.

Boston, Mass.

[While sprinkler systems are highly commendable and do lower insurance rates, they do not make unnecessary the presence of the factory fire chief. We did not mention these things in our editorial, for as stated in an editorial remark following a discussion letter commenting on our editorial on this same subject in the Dec. 31, 1912, issue, our field is not fire prevention. Our aim is to stop the practice of adding to the responsibilities of the engineer by appointing him chief of the factory fire department.—EDITOR.]

16 Lb. per Hp.-Hr., Noncondensing

I have read with interest the report in the July 15 number, page 105, of the test by F. W. Dean, of Boston, on an Erie City Iron Works' Lentz simple noncondensing engine. The results as claimed in the report are such as to cause some skepticism on the part of the reader.

There are a number of statements in the report as published, as well as a number of conditions which are not stated, that require further explanation before one can give full credence to the published results. I notice the statement: "Steam was so perfectly condensed that only a trifling quantity of vapor escaped, so small that it would not affect the result by a measurable quantity." If, as here admitted, the condensed steam left the condenser at so high a temperature as to steam in the open air, does this not vitiate the results?

It is known that the power of the atmosphere to absorb moisture is great; and if the water flowed from the condenser at a steaming temperature (which is admitted) the steam consumption of the engine would be some greater than that apparently shown.

It is common, in tests of boiler efficiency, to raise decided objection to the weighing in open tanks of feed water at steaming temperature; and there is no reason why the same argument should not prevail in an engine test.

I notice also the statement: "The reducing motion was practically correct." If the reducing motion is the same as usually employed on this engine, it may be practically correct, but not exactly correct; so is not this also an admission of possible error in the indicator diagrams by reason of a faulty reducing motion?

The article further states: "Current was absorbed by a water rheostat." I should like to know more regarding the kind of rheostat employed, and have further information regarding the steadiness of the load carried.

It is a difficult matter to carry an exactly steady load with a water rheostat, as usually constructed; and if, as is stated, "the indicator diagrams were taken every ten minutes, with a Crosby inside-spring indicator," it would be expected that the load was varying considerably during the ten-minute intervals; and it would be correspondingly quite impossible to determine whether the average of the indicator diagrams taken at so wide intervals equaled the average load as actually carried.

If the indicator diagrams were taken at moments when the water rheostat gave an excessive load, that would in part account, in addition to the other points above raised, for the apparently low steam consumption as claimed.

Nothing is said about the calibration of indicator springs; and here again is another uncertainty.

My further reason for questioning the accuracy of the results is that figures hitherto published, showing the steam consumption of this type and make of engine, do not show so remarkable results. I refer to POWER, June 3, where a 12x18-in. Erie City Lentz engine showed, with 149.6 lb. initial pressure, and 5 lb. back pressure, 25 lb. of steam per i.hp.-hr.; or a claimed result, by calculation for 150 lb. pressure and zero back pressure, of 22.25 lb. of dry steam per i.hp.-hr.

In POWER of Dec. 31, 1912, there is a report of a test on an Erie City Lentz 15x21-in. engine, at 145 lb. pressure, 2 lb. back pressure, 47 deg. F. superheat, in which the steam consumption per indicated horsepower-hour was shown to be 19.2 lb.

In both of these tests, as hitherto published, the engines were of a stroke which is long with respect to the bore, and in consequence a low clearance volume would be expected in both engines.

On the other hand, the engine recently tested is 19x21 in., and it is manifest that such a size of cylinder cannot have as small a percentage of clearance volume as in the case of the other two engines reported upon. Moreover, in this last test, the steam pressure carried is from 12 to 17 lb. lower than in the other tests.

The best result hitherto accomplished, as far as I have been able to learn, by test of a four-valve poppet-valve noncondensing engine is published in *Zeit. d. V. D. I.*, August, 1905, page 1310, where a 16.3x39.4-in. simple engine, with 145 lb. initial gage pressure, atmospheric exhaust, 254.3 deg. F. superheat, showed 16.10 lb. of steam per indicated horsepower-hour. Here is a poppet-valve engine with a stroke more than double the bore and, in consequence, with small clearance, and with superheat of over two and one-half times the amount of superheat carried on the Erie City Lentz test; yet this German engine shows the same steam consumption as claimed for this Erie City Lentz engine. Does it not appear probable, in the light of the above mentioned points, that the true results for the engine in question would be nearer 18 lb. of steam per indicated horsepower-hour?

ASA P. HYDE.

Binghamton, N. Y.

Hall of Records Test

Some misunderstanding is indicated in the editorial appearing in your issue of Aug. 26, upon the test now being conducted at the Hall of Records. The object of this test is to determine which is the more efficient method of obtaining light, heat and power for this building—by a private plant or from outside sources. The test is three cornered, in that it is being jointly conducted under the auspices of the administration of the Borough of Manhattan, the Bureau of Municipal Research and the New York Edison Co. Those in charge are of such unquestioned scientific standing and repute that the results, whatever they may be, will undoubtedly be accepted as final, so far as issues exist within the limits of the test.

In a part of your article, statement is made that "judging from past experiences, the representatives of the Edison Co. will undoubtedly try very hard to burden the plant with all sorts of fictitious fixed charges and it behooves the other members of the board to take a determined stand against such attempts." Would it be practicable to indicate an instance where this company has endeavored to burden any plant with fictitious fixed charges?

That such a test as this could be arranged with the officials of the city is gratifying evidence of the desire of those charged with the administration of the city's practical affairs to render the highest grade of public service. In this matter the truth only is sought, and whatever the result it cannot be harmful to the city, for if it be in favor of the continued operation of the plant assurance will then be had that no cheaper method of obtaining light, heat and power exists; if it be in favor of private electrical and steam service from outside sources, again the city will be benefited in securing the elimination of waste and needless expense, in that corresponding service can be obtained with less expenditure of money.

It should not be inferred that at this or any other time any criticism has existed concerning the conduct of this plant. Probably no private plant has ever been operated with greater watchfulness and efficiency, especially during the past two or three years under the present administration. It is a test of the best that can be done with a "private" plant, in comparison with the usual and ordinary commercial service of a corresponding nature.

ARTHUR WILLIAMS,
General Inspector,
N. Y. EDISON CO.

New York City.

ENGINEERS' STUDY COURSE

Boiler Efficiency

Solution of Practice Problem

At the plant in question the atmospheric pressure is 13 lb. per sq.in. and according to the steam table, the total heat of dry saturated steam at that pressure is 1148 B.t.u. per lb. and its temperature is 205.9 deg. Another reference to the steam table shows that the absolute pressure of steam having a temperature of 377.6 deg. is 190 lb. per sq.in., and the heat of the liquid of such steam is 350.4 B.t.u. per lb. while the latent heat is 846.9.

Then, substituting in the formula given last week for calculating the quality of steam we have

$$Q = \frac{(1148 - 350.4) + 0.46\ (286 - 205.9)}{846.9} =$$

$$\frac{797.6 + (0.46 \times 80.1)}{846.9} = \frac{797.6 + 36.8}{846.9} = 0.9852$$

or 98.52 per cent. as the quality of steam

Boiler Efficiency

Usually the sole purpose of a steam boiler is to generate steam. To do this, heat is required and this is obtained by burning fuel in a furnace connected with the boiler. In reality, there are two distinct steps in the process of steam generation and two parts in the apparatus employed. First, heat must be generated and for this the furnace and grate when solid fuel is used are required. Second, the heat generated must be transferred to water so as to convert it into steam and for this the boiler proper is provided. The degree of perfection or the efficiency with which each part does its work depends partly upon its design and partly upon the skill and care employed by the operators.

The efficiency of the boiler alone is the ratio of the heat absorbed by the water within to the heat supplied to the heating surface. To illustrate, if by calculation based upon the pounds of water evaporated and the pounds of fuel actually burned, it were found that for every 1000 B.t.u. actually liberated by the combustion of the fuel, 740 is transferred to or absorbed by the water in the boiler to make steam, while the balance was lost in various ways, the efficiency of the boiler would be as 740 is to 1000, or

$$\frac{740}{1000} \times 100 = 74\ \textit{per cent.}$$

On the other hand, the efficiency of the boiler and grate, taken as a whole, is the ratio of the heat absorbed by the water to the total available heat supplied to the furnace, the latter item being based on the number of pounds of fuel fired and the heat value of the fuel when fired.

The difference between these two methods of calculating efficiency is that in the first case the boiler is not charged with the loss due to unburned fuel dropping through the grate, which is proper as only the grate and not the boiler itself is responsible for loss in this direction. Hence, this loss is taken into consideration only when the boiler and grate are all considered as one unit.

It is customary to state both efficiencies when making a report of a test but the one of greater interest in everyday plant operation is the efficiency of boiler and grate. In brief form the two efficiencies are:

$$\textit{Efficiency of boiler} = \frac{\textit{heat absorbed per lb. fuel burned}}{\textit{heat value of 1 lb. fuel}}$$

$$\textit{Efficiency of boiler and grate} = \frac{\textit{heat absorbed per lb. fuel fired}}{\textit{heat value of 1 lb. fuel}}$$

A boiler test may be elaborate or simple, depending upon the object in view. With the elaborate test every factor possible is taken into consideration and the greatest possible accuracy is secured. With the simple test less effort is made and fewer factors are considered. The simplest test takes into consideration only the weight of the water apparently evaporated and the weight of the fuel supplied to the furnace. In between come the happy mediums which are elaborate enough to insure reasonable accuracy and simple enough to avoid expensive complications and justifiably dispensable effort. Such a one will be herein outlined. It may be further simplified on the judgment of the man who is conducting the test. All the principles involved and all the calculations entering into the making and reporting of this test have been taken up in detail in previous lectures and it only remains for the writer to show how they all are brought into application when conducting a test.

Test Apparatus Required

The following apparatus is required for properly making a test as here outlined:

1. Equipment for measuring or weighing water evaporated.
2. Equipment for measuring or weighing fuel.
3. Two draft gages; one at damper, one at furnace.
4. Four thermometers; one for feed water, one for steam calorimeter, one for flue gases and one for air.
5. One complete flue-gas analyzing set.
6. One steam calorimeter.
7. One outfit for making proximate fuel analyses.

Weighing Feed Water

As probably has been assumed from foregoing paragraphs, one of the main factors to be determined is the weight of water evaporated or steam generated during the test. This factor can be ascertained in two ways. First, the water can be measured or weighed before it is fed to the boiler or, second, the steam given off by the boiler can be collected, condensed and measured or weighed.

The latter method is, perhaps, the more reliable because only water that has been condensed from steam will be weighed while with the first method all the water pumped into the boiler may not be evaporated during the

test. If the blowoff should be leaky some of the water might escape in that direction without being converted into steam; also the water level in the boiler might not be the same in the end as in the beginning of the test, thus causing error in the amount of work credited to the boiler.

It is seldom, if ever, possible to condense and weigh the steam from a boiler being tested because usually it would be very inconvenient and expensive to install a special condensing equipment. Besides, this method would require taking the boiler out of service which sometimes cannot be done. However, with reasonable care the other method gives practically as reliable results.

Numerous methods of measuring or weighing the feed water can be devised and the one best to employ for any given test depends upon local conditions, size of the boiler, etc. Ordinarily, two tanks will suffice, one for measuring or weighing the water as it is received from the heater and one, to which the feed-pump suction is attached, to act as a receiving or storage tank. The tanks and the feed pump should be located as close as possible to the boiler to be tested so as to avoid radiation losses. The most convenient arrangement is to have the measuring or weighing tank placed upon a platform directly over the receiving tank so that the discharge to the latter will be as direct and rapid as possible. Preferably, the upper tank should rest on a set of scales so that each tankful may be actually weighed. If this is not convenient the tank may be provided with a suitable overflow and filled to overflowing each time, the capacity having first been accurately determined by either actually measuring or weighing the contents.

If possible, the thermometer for measuring the temperature of the feed water should be inserted in the feed pipe close to the boiler by means of a mercury thermometer well. If this cannot be done, the thermometer should be hung in the suction tank.

For measuring the fuel, if it is oil, a meter is usually employed, although a tank arrangement similar to that for the feed water could be used, suitable precautions being provided, of course, against fire. If the fuel is solid, two or more ordinary wheelbarows of suitable capacity will be required together with a set of ordinary platform scales. If the scales are of the portable variety, wooden runs for rolling the barrows up onto the platform and down off of it again will be found convenient.

The two draft gages, specified as item 3, while not absolutely essential, will be found instructive and helpful in analyzing results. One should be connected with the furnace and the other, with the uptake on the boiler side of the damper. Any type of gage will suffice, although the more accurate it is and the closer it can be read, the better.

The only absolutely necessary thermometer specified in item 4, is that for measuring the feed-water temperature. The others may be dispensed with if small economy is any object. In this case, the steam calorimeter would also have to be dispensed with and the quality of the steam, and the temperature of air and flue gases, all of which factors influence the results, would have to be guessed at.

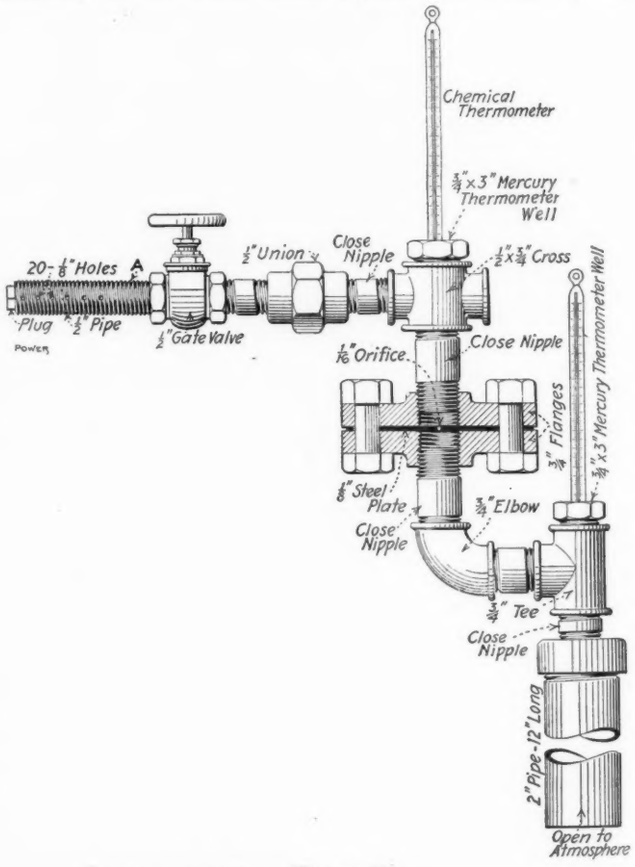
The cost of the thermometer for the feed water should not be very great as a range up to 212 deg. will do, because the feed water must be handled at atmospheric pressure and, hence, a temperature higher than 212 would be impossible. Thermometers for measuring the ordinary atmospheric temperatures are very cheap. Indeed, they are often given away as advertisements or souvenirs, consequently, there is little excuse for not using one during a test. It should be hung in such a location that it will indicate with fair reliability the average temperature of the air entering the boiler room.

The specifications and directions for using the thermometers for the flue gases and the steam calorimeter were given fully in previous lessons (July 29 and Sept. 16, issues) and need not be repeated here.

The function and value of the proximate coal-analysis outfit, flue-gas analysis apparatus and the steam calorimeter have also been fully discussed in previous lessons (May 20, June 17 and Sept. 16 issues). If a test is an important one it is advisable to send a carefully collected sample (see lesson in May 20 issue) of the fuel to a commercial chemical laboratory for proximate analysis and heat-value test in order to have a check on your own work.

Object of Test

Boiler tests may be made for almost any purpose: To test the performance of a new installation; ascertain the efficiency of the boiler under operating conditions; determine the maximum capacity that can be developed; find the value of a certain grade of fuel or to compare one grade with another; discover the effect of change in furnace design, type of grate or stoker, arrangement of baffles, etc.



Construction of Home-Made Steam Calorimeter

When tests for comparing one set of conditions or one

kind of fuel with another are made, especial care should be employed to keep the operating conditions as uniform as possible so that a perfectly fair comparison may be made. For instance, suppose a new furnace is being tried out. First, a test or series of tests with the old furnace would probably be run to discover the existing economy and then a test or series with the new equipment would be conducted to discover what improvement, if any, had been secured. Now, it would obviously be unfair to overhaul the boiler between the first and second tests or series, cleaning it thoroughly both outside and in and carefully pointing up all cracks in the setting, etc., and then credit to the new furnace the improvement in efficiency thus brought about. In the same way, it would be unfair to employ a higher grade of skill in operation during one set of tests than during the other, unless, of course, the men of the greater skill were to operate the plant continuously in the future.

On the other hand, if the boiler were to be tested to find out just what degree of efficiency was attainable, it would be poor judgment not to take every precaution possible in the way of tuning up just before the test.

Thus, the object or purpose of a test should be kept carefully in mind and the method suitably adjusted to attain the desired object as fairly and satisfactorily as possible.

[NOTE—The figure accompanying this lesson and illustrating the construction of a home-made steam calorimeter should have appeared with last week's lesson. Through an oversight in making up it was omitted.—EDITOR.]

Transporting Coal through Pipe Lines

The method of transporting coal by water pressure through a pipe has been suggested by E. G. Bell, engineer for the Hammersmith Electricity Works of London. By this method from 30 to 60 tons of coal per hour can, it is claimed, be forced through an 8-in. pipe leading from the Thames and under Chancellors' Road to the yard of the power plant on Fulham Palace Road, a distance of 600 yards.

If the scheme is adopted, a large mixing tank will be erected beside the wharf. The coal will be lifted from the barge by a mechanical grab, and after passing through an automatic weighing machine, it will be filtered into the mixing tank, which will contain approximately 15 per cent. of coal and 85 per cent. of water. A powerful electrically driven pump will then force the mixture of coal and water at high velocity through the transmission tube to a reception tank at the works.

The coal will sink to the bottom of this tank and another pump will drive the water through a return pipe to the mixing tank. Another grab will lift the coal out of the reception tank into the storage yard.

Although the initial cost of such a plant might be high, it is believed that it would eventually be a paying investment, since the present cost of transportation is approximately 14c. per ton, or somewhat over $3500 for the 25,000 tons which are used annually.—*Coal Age.*

The Indiana Coal Production in 1912, according to E. W. Parker of the United States Geological Survey, was 15,285,718 short tons. The increase over 1911 was 7.7 per cent. in quantity and 14 per cent. in value.

OVER THE SPILLWAY

JUST JESTS, JABS, JOSHES AND JUMBLES

Sure we were there! And you? If all the good things that were said and done were bunched, the answer would be: The Springfield N. A. S. E. convention was the best ever! And this is no idle boast, either. Everybody said so.

We met hosts of old friends and fraternized with lashings of new ones, to each of whom we would like to say something pleasant. But how can we in just one column of "Power"? The report of the convention is on page 450.

The only partiality we show is forced on us by the great work done by Walter Damon, the man who put the spring in Springfield. Indefatigable in his labors as chairman of the local committee to make the convention surpass all others, he worked day and night smoothing out the rough places and setting the machinery in motion. And he got away with it, always with a cheery word and an eye for the comfort of all.

Just one more word: We are tickled to the quick over the warm reception given our convention daily, the "Power Pup," and we thank you, one and all

The blacksnake whip has been abolished in an Indiana coal mine as a persuader of the mine mule. Looks like the only method left was to cuss 'em out. The mule that will locomote without either of these old-fashioned but heretofore effective persuasions must verily be of a new breed.

A centenarian gives advice to those of us who would hope to live 100 years. Among his rules are: "Don't worry. Keep out of jail." With some people, getting into jail has brought most of their worries to an end, but we can't all live in a big village like New York.

Don't get peeved when you read the article on the Lemp Brewing Co.'s plant in this issue. Nothing is said of its product. It's a good description of an efficient, well kept plant. Any good brewers' journal will give you the other details.

We agree with a former President that a pension list is a roll of honor. The Pennsylvania System has 3972 pensioners on its list. The road is proud of them, and the men are equally proud of their long-service records and the recognition thus given. How many years have you been with your company? How long will it be before you are entitled to a pension? There is a good, big thought in these two inquiries.

"How did I get to the top of the heap?" said a big executive.

"Why, I answered the questions before they were asked."

You get that, don't you?

"If you want to continue wearing your straw hat," says a contemporary, "and anyone tries to haul it down, spot him on the snoot!"

This is not a specimen of that editorial dignity with which we operate our editorial columns, but, taken unawares, we might say just this thing ourselves.

Thomas A. Edison's recent indisposition is ascribed to his taking a vacation, thus breaking up his long years of intense application and burning the midnight oil. Now you know why you are so healthy in your seven-day-a-week job.

Secretary of State Bryan in greeting the members of the International Congress of Refrigeration informed them that the principle of cold storage might well be applied to diplomatic relations.

"Diplomacy is the art of keeping cool," said Secretary Bryan. "While we do not desire to keep our foreign relations in cold storage, we want to refrigerate our disputes with other nations." He also said that such treaties as the peace agreements he proposed to the nations of the world on behalf of President Wilson will serve to keep the temper of nations cool, just as refrigeration keeps foodstuffs cool.

Throttling Calorimeters

SYNOPSIS—The principle of the throttling calorimeter is explained and illustrated. An example is given whereby the percentage of moisture in steam as determined by the calorimeter is worked out.

Percentage of moisture in steam below 3 per cent. can be determined by the throttling calorimeter, an apparatus in which steam is allowed to expand from pressures above atmosphere through a small opening and, by reading the pressures and temperature of the steam before and after expansion, the percentage of moisture can be determined.

The principle of the throttling calorimeter is best explained by reference to Fig. 1. Steam under pressure flows through the pipe at A and passes through the orifice B where it expands to atmospheric pressure, or nearly so, in the open pipe C.

Assuming that the contents at A is dry-saturated steam containing a certain number of heat units per pound, then dry-saturated steam at C would not contain the same number of heat units as when under pressure at A, and it

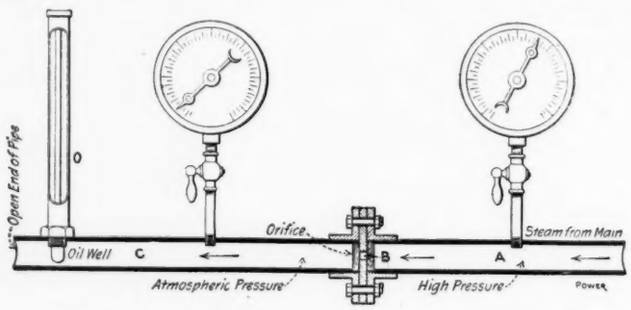


FIG. 1. DIAGRAMMATIC VIEW OF CALORIMETERS

follows that some of the heat in the steam at A will go toward superheating the steam at C at the lower pressure. If any moisture is present when the steam is at A, some of the heat in the steam will go toward evaporating this moisture when the steam expands at the orifice. It follows that the temperature of the steam will then not be as high as when there was no moisture.

For example, suppose steam at A is at 90 lb. absolute pressure, dry saturated, then by referring to a steam table it can be found that the total heat (above 32 deg. F.) in 1 lb. of steam will be 1184.4 B.t.u. At atmospheric pressure 1 lb. of steam when dry saturated contains a total heat (above 32 deg. F.) of 1150.4 B.t.u. From this it is seen that

$$1184.4 - 1150.4 = 34 \textit{ B.t.u.}$$

are available and that this must go toward superheating the steam at atmospheric pressure. Then 34 B.t.u. will give

$$34 \div 0.48 = 71 \textit{ deg. of superheat}$$

the specific heat of superheated steam at atmospheric pressure being say 0.48. As the temperature of dry-saturated steam at atmospheric pressure is 212 deg. F., there would be a thermometer reading at zero of

$$212 + 71 = 283 \textit{ deg. F.}$$

Now if the steam at A, while still at 90 lb. absolute pressure, had contained a percentage of moisture, some of the available 34 B.t.u. would have gone to evaporate this moisture and only a part would have superheated the steam, so that the thermometer reading would have been less than 283 deg. F. This thermometer reading serves to determine the percentage of moisture present in the steam at A. From this it can be seen that if enough moisture is present at A to take up all the 34 B.t.u. to evaporate it so that there will be no superheating—that the thermometer will not register over 212 deg. F.—the limit of the usefulness of the throttling calorimeter is arrived at and, for this reason, it cannot be used for moisture of over 3 per cent.

The formula for the percentage of moisture in steam as determined by the throttling calorimeter is

$$W = 100 \times \frac{H - h - K\,(T - t)}{L}$$

where

W = Percentage of moisture in the steam at A;
H = Total heat in the steam at the high pressure;
h = Total heat due to pressure at C, which is 1150.3 B.t.u. at atmospheric pressure;
K = Specific heat of superheated steam = 0.48;
T = Temperature of the throttled or superheated steam at C;
t = Temperature due to pressure at C, which is 212 deg. F. at atmospheric pressure.

H, L, H and t can be found in the steam table when the pressures are known.

To illustrate, assume that the pressure at A is 75 lb., that at C is at atmospheric pressure, or 14.7 lb., and that the thermometer reads 232 deg. F. Then by turning to the steam table

$H = 1181.1$;
$L = 893.9$;
$h = 1150.3$;
$t = 212$ deg. F.

It is assumed that

$T = 232$ deg.;
$K = 0.48$.

Then substituting in the formula

$$W = 100 \times \frac{(1181.1 - 1150.3) - 0.48\,(232 - 212)}{893.9}$$

$$W = 100 \times \frac{30.2 - 0.48 \times 20}{893.9} = 100 \times \frac{30.2 - 9.6}{893.9}$$

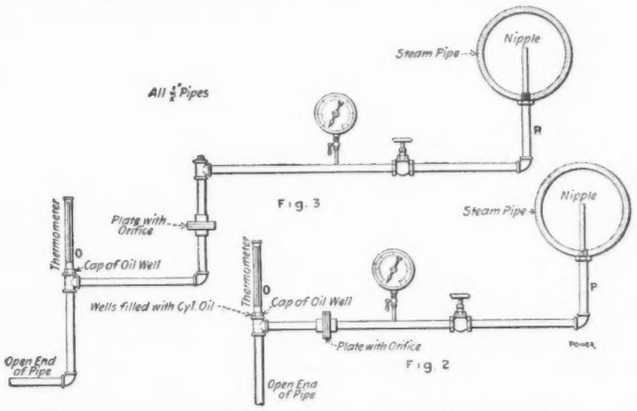
$$= 100 \times \frac{20.2}{893.9} = 100 \times 0.022 = 2.6$$

or 2.26 per cent. of moisture in the steam at A.

How to Make a Calorimeter

An engineer without a calorimeter and not caring to buy one, can make it. Figs. 2 and 3 show two forms of calorimeters. Use ½-in. pipe throughout and cut it to any reasonable and convenient length. The nipples inserted into the steam pipe or union are open only at their upper ends and should reach to the center of the steam pipe, but if preferred, the sides can be perforated with a few holes not over ⅛ in. in diameter. The nipple is essential because if it were only screwed and did not protrude into the central part of the pipe, the condensation would pass into the calorimeter and give erroneous

results. The gages are not essential, as the pressure in the pipe is probably known from gages elsewhere in the line, but it is advisable to have one on or near the calorimeter for convenience and accuracy. The thermometer is necessary, but it must be accurate, with a range from 200 to 350 deg. F. The plate held in the flange union should be about ⅛ to ¼ in. thick, and have a hole in the center $\frac{1}{16}$ to ⅛ in. in diameter. The entire length of the apparatus should be covered to prevent condensation as far as possible. Only the valve wheel should be free and a hole left in the covering for the thermometer. The oil well at *O* should have a cup whose outside diameter is not large enough to allow steam to pass around it freely in escaping to the atmosphere.



FIGS. 2 AND 3. VERTICALLY AND HORIZONTALLY CONNECTED CALORIMETERS

When taking readings the valve should be opened full and steam allowed to flow for a few moments to heat the apparatus. When the thermometer shows a constant temperature the reading should be taken. This reading represents *T* in the formula. The pressure on the high side should also be observed from which the values *H* and *L* are found in the steam tables. As the low pressure is assumed to be atmospheric, $h = 1150.4$ and $t = 212$ deg. F.

INSTRUMENT ERROR

In the preceding example it was assumed that the temperature of the steam on the low-pressure side would be 212 deg. F. if the steam were dry saturated, and if the steam on the high-pressure side were dry saturated at 90 lb. pressure the temperature of the superheated steam in the low-pressure side, as registered by the thermometer, would be 283 deg. F. This is known as the "normal reading." But the thermometer will not register 283 deg. even if the above conditions prevailed as the reading will be slightly lower.

One reason is that a certain amount of heat will radiate through the calorimeter walls even if it is covered, and another is that the stem of the thermometer is exposed to the temperature of the surrounding air. Both of these sources of error will tend to give a temperature reading lower than it should be. This reading is represented by *T* in the formula. The difference between the reading as it should be and that which is actually the case is known as the error of the instrument.

It is well, therefore, to determine what the error of the instrument is and to make a chart of corrections on section paper, as shown in Fig. 4. Let it be assumed that the steam plant is not in operation and that the steam present in the boilers and piping is dry or "dead" steam. If the steam is dry, it can be calculated what the normal gage readings should be, by subtracting the temperature at *A* from that at *C*, which will be zero, or atmospheric pressure, and divide the remainder by the specific heat of superheated steam at atmospheric pressure, or 0.48, and add the result to 212, the temperature of steam at atmospheric pressure. For the following pressure the normal readings are:

Pounds, gage	Degrees F.
20	246.4
40	264.3
60	276.1
80	285.1
100	292.2

Now take a piece of sectional paper and under the lower line lay off values for steam temperatures. Then plot the foregoing values and draw the "curve of normal readings" through these points as shown.

At some convenient time when the steam is "dead" and supposedly dry saturated, regulate the pressure in the pipe to the values referred to; that is, 20, 40, 60, 80, 100 lb. Then turn steam into the calorimeter and note the temperature registered by the thermometer after the steam has flowed for about four minutes and is constant. It will be seen for each pressure that the values will be lower than the calculated ones. For example, assume that these values would be

Pounds, gage	Degrees F.
20	230
40	243
60	251
80	256
100	262

With these values plot the second curve or "curve of actual readings." This will give a chart showing the error of the instrument for all temperatures likely to occur. Suppose that the calorimeter is in actual use and that the percentage of moisture in steam is being determined, the thermometer shows 240 deg. F. This is the actual reading and lower than the normal reading in the formula.

Referring to the chart, a point is found on the lower curve corresponding to this value, and directly above it on the normal reading line is the value 261 deg. Then

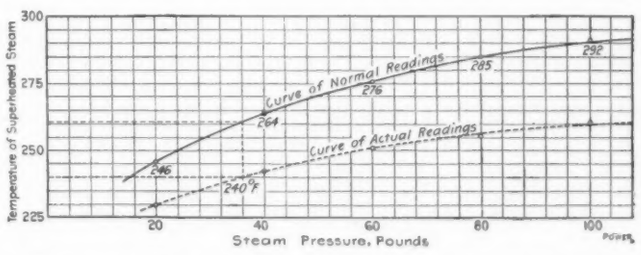


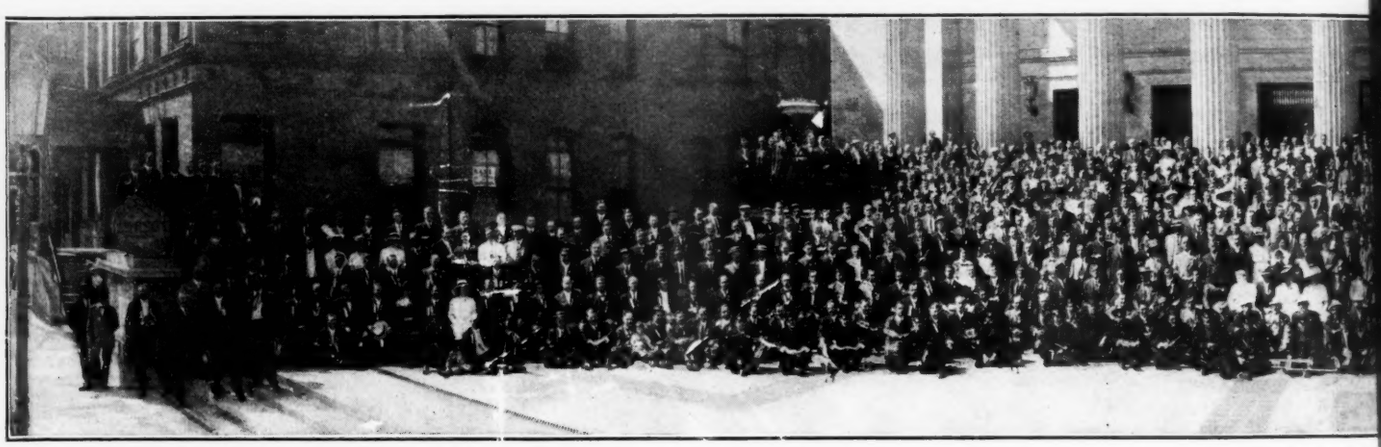
FIG. 4. CURVES OF NORMAL AND ACTUAL READINGS

when working out the formula used for *T*, not the actual reading 240 deg. F., but the other, 261 deg. F. This will give the corrected temperature reading and take care of all radiation and other losses of heat in the calorimeter which will be close enough for general purposes. It should be remembered that the figures assumed for the curve of actual readings are given only as an example. Each calorimeter will give different values and these must be determined for each instrument. The values worked out for the curve of normal readings should be used.

The N. A. S. E. Convention

Ideal convention weather, an ideal convention city and a large and enthusiastic attendance, including 412 accredited delegates, combined to make the thirty-first annual gathering of the National Association of Stationary Engineers at Springfield, Mass., one of the most successful in the history of the organization. In a city like Springfield the convention is not absorbed as it is in a metropolis. The people know that it is there and enter into the spirit of it and the mayor does not send an understudy but comes himself to welcome the visitors and takes a human interest in their presence. At Springfield there were present not only the mayor of the city itself but those of the neighboring cities of Lawrence and Northampton.

The visitors were gracefully welcomed by Mayor Denison, and speeches were made by representatives of the Board of Trade, the Publicity Club and the mayor of Lawrence, which were fittingly responded to by President McGrath, Vice-President Coe and Past President Kearney. The local speakers showed in their remarks an unusual degree of familiarity with the objects and methods of the association and of acknowledgment of its high mission and practical importance as an industrial factor.



DELEGATES AND GUESTS AT THE NATIONAL ASSOCIATION OF STATIONARY

The president's annual report commended the work of the educational committee, condemned carping criticism of the methods of the association and recommended the continuance of the present program until something definitely better is developed.

In order to secure better efficiency and more conformity in the work of the state associations he recommended that the programs of their annual conventions be submitted to the national president in advance for approval, which recommendation was adopted. A suggestion that the reports of the subordinate associations be sent directly to the national secretary instead of to the deputies was defeated.

The secretary's report showed the membership to be something over 21,000, and the treasurer's report showed a balance in favor of the association of over $27,000, of which upward of $18,000 was required to pay the mileage of the delegates.

The Life and Accident Association shows a membership of 3587, an increase of 346 over that of last year. Its disbursements during the year, outside of benefits paid, have been less than one dollar per member and it has over $5000 in reserve. J. D. Taylor was reëlected secretary-treasurer and J. G. Beckerleg and T. N. Kelsey as trustees.

In response to 340 letters sent out by a committee appointed by resolution of the last convention "to consider the matter of biennial conventions of this national body and report at the next annual meeting," only 162 replies were received, of which 116 voted against biennial and 45 in favor of the two-year plan with one tie vote. The committee simply reported these facts without recommendation.

A committee appointed to consider the advisability of publishing a history of the association, reported that "in the office of the national secretary there is a continuous record of convention proceedings from which might be compiled a digest of the legislative work of the N. A. S. E., which would be of value as a work of reference." They did not favor the writing of a history for the reasons "that the association has not been in existence long enough to warrant a written history; that there is no real demand for such a history; that it would require a considerable sum of money to publish a history, the sale of which is a matter of mere guesswork."

The president decided that the New England States Association had no legal standing, and was supported by the convention. A resolution authorizing the organization of district associations was defeated.

The License Committee reported in detail the efforts which had been made to secure legislation providing for the examination and licensing of engineers during the past year. The net result was the passing of the New Jersey law, of which we have already treated at length, without any appropriation to make it effective; an amendment to the Pennsylvania law, making it mandatory upon all cities of the third class to pass ordinances requiring the licensing of steam engineers, and to create departments of boiler inspection; a state law in California similar to that of New Jersey, and ordinances in Geneva, N. Y., and Decatur, Ill. The convention adopted a recommendation of the Ways and Means Committee that an amount not to exceed $2000 be set aside to assist cities

and states in securing license legislation during the coming year. The compilation and publication in book form of all existing engineers' license and boiler-inspection laws and ordinances was authorized.

The trustees of the *National Engineer*, the organ of the association, reported the paper in flourishing condition, having netted the association over $17,000 the past year. They protested against the publication of the directory of officers and associations, because it afforded to those who might otherwise patronize the paper a means of reaching the membership without doing so. They declared that "the practice of soliciting so called donations from our patrons, which is in reality a form of advertising by the exhibition of business cards in the association rooms, picnic tickets, etc., is bringing to us adverse criticism from many quarters" and strongly urged the discontinuance of this practice. They recommended that a sum not to exceed 25 per cent. of the net income of the paper be rebated *pro rata* as per capita tax paid, and perhaps the most important act of the convention was to adopt this recommendation.

ENGINEERS' CONVENTION AT SPRINGFIELD, MASS., SEPT. 8 TO 13, 1913

During the year 182 members of the association have passed away and the delegates stood in impressive silence while their names were read by the secretary.

A feature of the convention was an address by Edward F. Miller, professor of mechanical engineering at the Massachusetts Institute of Technology and leading member of the Advisory Educational Board of the N. A. S. E. He traced the development of the civil from the military engineer, the growth of power as a factor in industry and civilization and of the men concerned with its production.

J. D. Taylor told of the efforts which were being made in New York to secure uniformity in rates for electric service and prevent the sale of current at less than cost to put isolated plants out of business. George H. Diman, of Lawrence, Mass., was accorded the floor as an honorary member and advised the delegates to work for a flat rate and the prevention of discrimination, which was of more moment to them than license laws, with which he has little sympathy.

The Educational Committee reported that 33 members had competed in the first or December to February Prize Question Contest, 20 in the advanced and 13 in the elementary. The prizes were awarded as follows:

ADVANCED CONTEST

First, $75—Frederick L. Ray, Kentucky, No. 3.
Second, $50—William T. Frizzell, Massachusetts, No. 16.
Third, $30—H. E. Pressinger, Wisconsin, No. 18.
Honorable mention—A. Wade, Illinois, No. 28.

ELEMENTARY CONTEST

First, $50—Charles E. McLean, Lowell, No. 17.
Second, $30—John O. Burman, Lowell, No. 17.
Third, $20—Norman S. Campbell, Michigan, No. 1.

In the second or March to June contest there were 18 papers submitted, 7 in the elementary and 11 in the advanced. The winners were:

ADVANCED COURSE

First, $75—R. B. Ambrose, Pennsylvania, No. 3.
Second, $50—H. E. Pressinger, Wisconsin, No. 18.
Third, $30—Martin O'Connor, Wisconsin, No. 1.

ELEMENTARY COURSE

First, $50—C. B. Hudson, Massachusetts, No. 17.
Second, $30—John Guilfoy, Massachusetts, No. 16.
Third, $20—divided equally between J. O. Burman and C. E. McLean, of Massachusetts, No. 17.

A second activity of the committee was to offer four prizes for meritorious work in association meetings. The conditions were that the meeting should be of educational nature, that the members should take part in the discussion and that a report of the number present should be sent to the Educational Committee by May 1. "This competition," says the committee, "was advertised twice, but apparently it was not received very enthusiastically by the associations. At any rate there were only three entries—two of these from the same association. The prizes announced were to be of equal value with no mention of first, second, etc., and the original announcement was to the effect that the prizes would probably consist of steam-engine indicator outfits. The best entry was that of Illinois No. 1 for a meeting held on Jan. 15, 1913, when A. R. Maujer, of Illinois No. 1, gave a lecture on 'Proximate Coal Analysis and Its Value in Power-Plant Economy.' Judging from the report received from the officers this meeting was a most successful one—139 members and friends were present, and 21 took part in the discussion which followed.

"The second meetings in order of merit were those of Kentucky No. 1. This association reported two meetings, one on Nov. 27, 1912, and another on Mar. 12, 1913. At

the meeting of Nov. 27, D. L. Fagnan, of the De La Vergne Co., lectured on 'Oil Engines,' before an audience of 58 men. The discussion was mainly in the form of questions and was not in the same class as that of Illinois No. 1; nevertheless, we recommend the awarding of a prize to this association and it will be left to the national president to make both awards."

The committee reported that the Information Department, which acts in a consulting capacity for solving plant problems presented by the members, had found much favor. About 75 problems were presented to the department throughout the year.

Another successful innovation was the distribution of lantern slides to be used by local members in lectures. Sets of slides dealing with turbine, boiler and gas engine have been distributed and one on refrigeration is in preparation. When this is finished the association will have 2550 lecture slides. The committee recommended the continuation and extension of the lantern-slide collection and system of lectures and of the Information Department, the continuation of the offering of prizes for answers to questions for another year; a subdivision of lantern-slide sets; the continued award of prizes for well attended educational meetings, the discussion to be a factor in making the award; the urging of state associations to take up the work offered by the National Educational Committee. A sum not to exceed $3000 was appropriated for these purposes.

Mesdames Heckler and Mullen, president and past president of the Ladies Auxiliary, visited the convention, described the work which they were doing and made a plea for continued support, especially of the pension department, which had collected more than $2100 and disbursed more than $1500 to the pensioners during the past year.

The newly elected officers of the auxiliary are: Mrs. Sophie Whalen, of Chicago, president; Mrs. Hunt, of Boston, vice-president; Mrs. Frank Flynn, of Chicago, secretary; Mrs. Herbert E. Stone, of Framingham, treasurer; Mrs. Huntington, conductor.

Of eight cities which wanted the convention next year, Milwaukee was selected. This will be the third time, the last being in 1900.

The officers elected for the ensuing year are James R. Coe, of New York, president; Frederick L. Ray, of Louisville, Ky., vice-president; Fred W. Raven, of Chicago, secretary; Samuel B. Forse, of Pittsburgh, treasurer; Joseph F. Carney, of New York, trustee of *National Engineer;* John J. Reddy, of Jersey City, conductor; John P. Proper, of Los Angeles, doorkeeper.

The social features comprised a reception by the national officers on Monday evening, a visit to the works of the Chapman Valve Co., a gigantic clam bake, at which over 1900 people were served, a smoker, a vaudeville performance, afternoon concerts and teas and a grand ball following upon a public installation of the officers.

The exhibition conducted by the National Exhibitors' Association in connection with the convention occupied two floors of the large municipal auditorium and was well attended by local engineers as well as by the visitors.

There was an unusual number of operative exhibits which attracted particular attention, such as two full-sized Corliss engine models operated by motors, traps and high- and low-water alarms in operation, a device for testing the relative friction of packing on 3-in. rods, autogenous welding, cutting, etc. A local welding company subjected a 30-in. vertical firebox tubular boiler made of ¼-in. stock entirely by welding to hydrostatic pressure and it developed a leak in the welded-in door frame at 305 lb., due to the inward bulging of the unstayed firebox. As the boiler was designed for heating service under 10 lb. pressure, the test was regarded as very unsatisfactory.

The newly elected officers of the Exhibitors' Association are Homer Whelpley, of Cleveland, president; Henry Pastre, of Pittsburgh, vice-president; Lane Thompson, of Cincinnati, secretary; Harry D. Raymond, of Boston, treasurer; Charles Cullen, of Cincinnati, Charles A. Wilhoft, of New York, Paul T. Payne, of Chicago, Joseph H. Meyers, of Cleveland, and John H. Foote, of New York, members of the executive committee.

It goes without saying that retiring officers and those who had especially contributed to the success of the convention were remembered in resolutions of thanks and more substantial testimonials, including a handsome watch to Walter Damon, who had been untiring in his efforts to bring the convention to Springfield and as chairman of the local committee fulfilled all of the anticipations which he had aroused.

Determining Reducing-Valve Sizes

By J. Rowland Brown

A large percentage of the troubles experienced with steam-pressure regulating or reducing valves can be traced directly to the use of valves too large for the service conditions they are required to meet. It is common practice when installing a regulating valve in a heating system or on a cooking apparatus to use a valve of the same size as the pipe leading to the system. An exception to this practice is in vacuum-heating systems where it is customary to install a valve of one-half the diameter of the service pipe and have the outlet of the valve twice the size of the inlet of the port. This latter case, while a move in the right direction, may be in serious error.

The correct way to determine the size of a regulating valve is to find the weight of the steam required per hour and then choose the valve that will most nearly deliver that quantity of steam at the required pressures. The objection to this scheme is that it is often difficult or impossible to determine the weight of steam required.

The following is a description of a scientific method of approximating the correct size of valve, and, though only an approximation, it has proved, when tried in some hundreds of cases, to be seldom in error.

The upper curve, Fig. 1, shows the relation between the specific volume of saturated steam and its absolute pressure, and the lower curve represents the relation between the square root of the specific volume and the absolute pressure. It is readily seen from an inspection of the upper curve that the volume of a pound of steam is very great at low pressures and decreases very rapidly as the pressure rises up to about 75 lb. absolute and then decreases very slowly. As an example, 1 lb. of steam at 100 lb. absolute pressure will occupy 4.4 cu.ft.; if its pressure is reduced to 50 lb., it occupies 8.4 cu.ft.; to 25 lb. 16 cu.ft., and at atmospheric or 15 lb. absolute pressure it occupies 26 cu.ft.

The quantity of steam that should flow through a pipe

is generally figured by assuming some velocity of flow, and then knowing the pressure and specific volume, the weight per hour is calculated.

Let

A = Area in square feet of the pipe or passage;
V = Velocity of steam in feet per minute;
S = Specific volume of saturated steam in cubic feet per pound;
D = Diameter of pipe or passage in feet.

The pound of steam flowing through the pipe per minute is equal to

$$\frac{A \times V}{S} = \frac{0.7854\, D^2 \times V}{S}$$

In the case of a valve having the same size of inlet and outlet, the number of pounds of steam per minute leaving the valve is the same as the number entering, and there is the following relation:

$$\frac{A \times V}{S} = \frac{A \times V_1}{S_1} \text{ or } \frac{V}{S} = \frac{V_1}{S_1} \text{ or } \frac{V}{V_1} = \frac{S}{S_1}$$

In other words, the ratio of the inlet and outlet velocities is equal to the ratio of the inlet and outlet specific volumes. In a regulating valve having an equal size inlet and outlet this accounts for the fact that the velocity is higher in the service or outlet pipe than it is in the initial or inlet pipe. There is no economy in having the velocity in the inlet pipe lower than that in the service pipe, as a smaller inlet pipe and a smaller valve will do the work. The port of the valve should in all cases be the size of the inlet pipe, if the pipe is the correct size. The greatest economy exists when the velocity of steam is the same in the inlet and outlet provided the outlet velocity is correct.

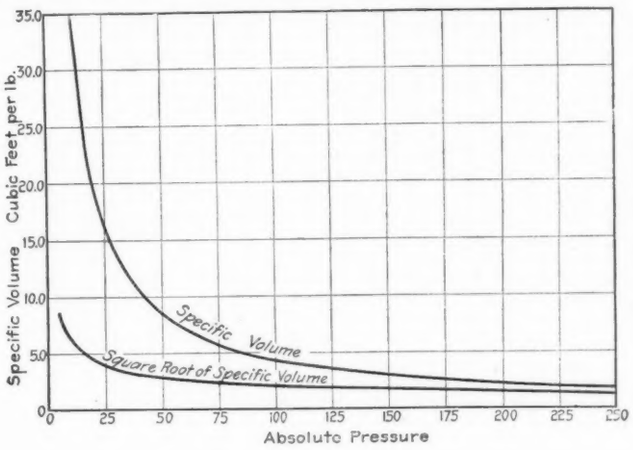


FIG. 1. PRESSURE-VOLUME CURVES

In the case of the inlet and outlet velocities being equal, the ratios are

$$\frac{A \times V}{S} = \frac{A_1 \times V}{S_1} \text{ or } \frac{A}{S} = \frac{A}{S_1} \text{ or } \frac{A}{A_1} = \frac{S}{S_1}$$

in other words, the ratio of the inlet and outlet areas is equal to the ratio of the inlet and outlet specific volumes.

To find the ratio of the diameters:

$$\frac{A}{A_1} = \frac{0.7854\, D^2}{0.7854\, D_1^2} = \frac{D^2}{D_1^2}$$

$$\frac{A}{A_1} = \frac{S}{S_1} = \frac{D^2}{D_1^2} \text{ or } \frac{D}{D_1} = \frac{\sqrt{S}}{\sqrt{S_1}} \text{ and } D = D_1 \frac{\sqrt{S}}{\sqrt{S_1}}$$

that is, the ratio of the diameters of the inlet and outlet pipes is equal to the ratio of the square roots of the specific volumes; or the diameter of the inlet pipe or the regulating-valve port is equal to the product of the diameter of the outlet and the ratio of the square roots of the specific volumes.

It is necessary, therefore, to know the size of the outlet pipe and the initial and service pressures, and by referring to the curve, Fig. 1, the square roots of the specific volume can be found.

The single assumption made in this problem is that the service or outlet pipe is of the correct size. This formula insures a valve of such size as to give the same steam velocity at its inlet and through an inlet pipe of the same size as that in the outlet or service pipe, and, therefore, provides a smaller valve than would ordinarily be selected. The use of the smaller valve is, of course, cheaper in first cost, and because of the main valve being wider open and there being less wire drawing or cutting of the seat, the valve should last longer. There is also a decided advantage in operation as the smaller valve will have a greater lift of disk for a given amount of steam flowing and is, therefore, not disturbed by vibration, sticking or pressure changes to as great an extent.

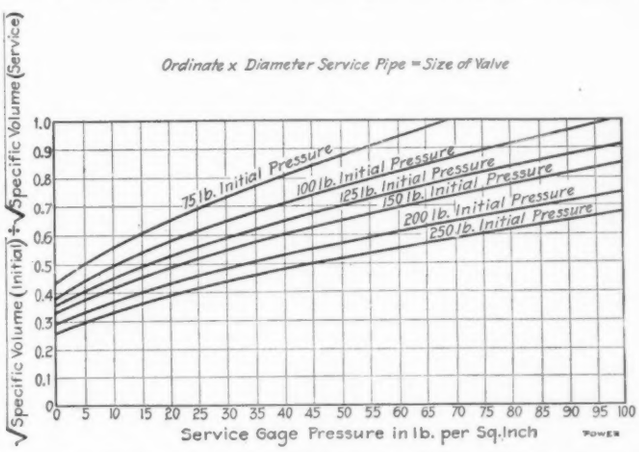


FIG. 2. CURVES FOR DETERMINING SIZES OF REDUCING VALVES FOR GIVEN SERVICE PRESSURES

It is difficult to get good, continuous regulation when a valve is just cracked off of its seat.

Referring to the lower curve, Fig. 1, and to the formula

$$D = D_1 \frac{\sqrt{S}}{\sqrt{S_1}}$$

take, as an example, a 12-in. service pipe with a pressure reduction of 145 lb. to 1 lb. gage pressure, or 160 lb. absolute to 16 lb. absolute.

$$D_1 = 12 \text{ in.} = 1 \text{ ft.} \quad \sqrt{S} = 1.7 \quad \sqrt{S_1} = 5.0$$

and $D = 4.08$, which is very close to a 4-in. valve, which would be the size to use.

To make the above operation easier, the curves, Fig. 2, were drawn, in which the ordinates represent the ratio $\frac{\sqrt{S}}{\sqrt{S_1}}$ and the abscissas the service pressure recorded by the gage (not absolute).

To use the curves, first find the service pressure at the bottom and follow the vertical line until it intersects the

curve of initial pressure; then follow the horizontal line through this intersection to the left and read the corresponding number. The number at the left multiplied by the size of the service pipe (in inches) will give the size of the inlet pipe and valve.

As an example, take the case of a 4-in. service pipe and a reduction of 75 lb. gage to 20 lb. Read 20 lb. at the bottom of the curves, then follow the vertical line until it intersects the 75-lb. curve, next follow horizontally to the left and the reading is 0.64. Multiply 0.64 by 4, the size of the service pipe, and the result, 2.56 in., is the size of the valve. The nearest commercial size of valve and inlet pipe would, of course, be 2½ in.

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Fire-Fighting Talks to Engineers

On Monday evening, Sept. 8, about 1800 factory fire chiefs, many, if not most of them, chief engineers, listened to instructive talks by Colonel Bryant, Commissioner of Labor, of New Jersey, and ex-Chief Croker, of the New York Fire Department, at the Krueger Auditorium, Newark, N. J. Several excellent lantern-slide illustrations of factory fire-fighting apparatus, such as pumps, etc., were shown.

NEW PUBLICATIONS

ELECTRICITY FOR THE FARM AND HOME—By Frank Koester. Published by Sturgis & Walton Co., New York, 1913, as part of the "Farmers' Practical Library." Size, 8x5 in., 279 pages, illustrated. Price, $1, net.

The application of electricity to farm work has been touched on from time to time in various magazines, but this is, insofar as we know, the first extensive treatment of the subject in book form. It is a book which should appeal to every progressive farmer.

The author points out the three possible sources of electricity for this purpose. First, high-tension transmission lines from large central stations often pass through farming country and these can very easily be tapped and the voltage stepped down through small transformer stations to supply one or a group of farms.

The second plan is to establish rural central stations. This practice has been followed to a large extent in Germany, such stations often supplying from 100 to 150 farms. Where rural central stations are not feasible and where no transmission lines pass in the vicinity, the farmer may, providing his work is extensive enough, put in an isolated plant. This is often advantageous where there is water power available on the premises. Typical installations along this line are cited.

A chapter is devoted to various motor applications in farm work, and rules are given for determining the sizes of motors to be used. Useful cost figures are also included.

The book concludes with suggestions for electric heating and lighting on the farm.

LAYING OUT FOR BOILER MAKERS AND SHEET METAL WORKERS—SECOND EDITION. Published by the Aldrich Publishing Co. ("The Boiler Maker"), New York. Cloth; 306 pages, 10x13 in.; 600 illustrations. Price $5 postpaid.

Those having a copy of the first edition will not need to be told that it is an excellent work on the subject, giving in a practical way all the information necessary to enable the boilermaker to lay out in detail different types of boiler, tank, stack and irregular sheet metal work. The first two chapters explain the methods of laying out by orthographic projection and triangulation and problems are given to illustrate the work. In the chapters dealing with the layout of different types of boiler more space is given to the plain tubular type as the problems are general and may be applied to almost any other type. Rules for determining the size, shape and strength of the different parts are also included. In the second edition 113 additional pages have been added, comprising 44 new laying out problems and chapters on miscellaneous calculations and tools for boiler makers. The problems cover a wide range of work, showing the layout and construction of regular and irregular elbows, pipe connections, transition and offset pieces, taper courses, spiral pipe, hemispherical water tanks, firebox wrapper sheets for locomotive boilers and smokestack collars, uptakes and smokeboxes for Scotch boilers. The miscellaneous calculations referred to, show how to figure the efficiency of riveted joints, the area of circular segments and the cost of boiler construction. For simplicity and breadth of treatment, anyone interested in the work outlined must need go far to find a better book on the subject.

STEAM ENGINEERING—A Text Book. By William B. King, Published by John Wiley & Sons, New York City and London. Cloth; 439 pages, 6x9 in.; 177 illustrations. Price, $4, net.

The book is a simple and straightforward treatment of the principles and action of steam boilers, reciprocating engines, turbines and the accessories, requiring no extensive knowledge of physics or thermodynamics. With so broad a field the treatment must naturally be brief, but is by no means too much condensed for a ready and clear understanding of the subjects discussed. The thermodynamics and kinematics involved are made so simple that previous knowledge of the subjects is not necessary. The table of contents following will give some idea of the compass of the book: Chapter I, Introductory; Chapter II, Application of Heat to Water; Chapter III, Fuels and Combustion; Chapter IV, Steam Boilers, Attachments and Accessories; Chapter V, The Slide Valve and Its Motion; Chapter VI, Rotary Motion, Action of Crank and Connecting Rod; Chapter VII, The Reciprocating Steam Engine, Types and Details; Chapter VIII, Condensers, Feed-water Heaters, Pumps; Chapter IX, The Indicator and Its Diagram, The Planimeter, Clearance, Ratio of Expansion; Chapter X, Expansion of Gases, Mean Pressure of an Expanding Gas; Chapter XI, Compound or Stage-Expansion Engines; Chapter XII, Boiler Efficiency, Calorimetry; Chapter XIII, Engine Efficiency, The Carnot Cycle; Chapter XIV, Design of Simple and Compound Engines; Chapter XV, Combining Diagrams of Stage-Expansion Engines; Chapter XVI, The Zeuner Valve Diagram; Chapter XVII, Entropy; Chapter XVIII, The Steam Turbine; Chapter XIX, Testing Engines and Boilers; Index.

Of superheated steam the specific heat is no longer taken at the constant value of 0.48. Recent experiments have shown that it varies with the temperature and the pressure. The author has used the older value and in the first two chapters values from the older steam tables, although in the balance of the book the Marks and Davis values have been adopted.

It is, of course, best to be uptodate and consistent on data of this character. Aside from this the book is the best we have seen in many a day. It is simple and explicit, and gives what is wanted in a nutshell. It should be an excellent work for a short course or for an introduction to subjects to be taken up more in detail later on. As a reference to refresh the memory and give all that is necessary in a short space, a better book could not be obtained.

SOCIETY NOTES

The evening courses offered by Pratt Institute will open on Oct. 1. The courses are technical chemistry, mechanical drawing, industrial electricity, practical electricity, steam engine, strength of materials, practical matnematics, machine work and tool making, carpentry and building, pattern making, sheet-metal work, plumbing, trade teaching. A circular of information may be had by writing to School of Science and Technology (Dept. N), Pratt Institute, Brooklyn, N. Y.

PERSONALS

C. F. Catillaz, the New York representative of Neemes Brothers, has moved into his new office at 30 Church St., New York City.

Henry D. Jackson, consulting engineer, Boston, Mass., sailed Sept. 20 for Bermuda on a three-months' trip for rest and recreation.

R. A. Foresman has been appointed supervising engineer of the Eastern Pennsylvania Power Co. plants, with headquarters at Easton, Penn.

William Tietze, formerly chief engineer of the Power Mining & Machine Co., Cudahy, Wis., and now with the Pierce Phosphate Co., Pierce, Fla., has been appointed chief engineer with this company.